THE DESIGN AND CONSTRUCTION

OF

STEAM TURBINES.

A MANUAL FOR THE ENGINEER.

BY

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AND THE METHOD OF LEAST WORK."

With Numerous Reproductions from Working Drawings.

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PREFACE.

THE theory of the steam turbine, as developed in this volume, is based upon articles contributed by the author, during the past few years, to the columns of ENGINEERING. To the Editors of this journal he is deeply indebted for permission to reproduce what he desired of the text of these, and for authority to reprint a selection from the series of working drawings of turbines, which they have published during the same period.

The descriptive matter which accompanies these illustrations is slightly condensed from the original articles. Of these the great majority were written by the author, but his former colleague, Mr. R. H. Parsons, A.M.I.C.E., was responsible for two or three, and in particular for the detailed and singularly clear account of the Ljungström Steam Turbine. The author has also to thank Mr. Parsons for much valuable and early information as to changes in turbine practice, and for much helpful criticism.

Mr. Alexander Richardson procured for the author complete data of important turbine tests, which have proved invaluable.

To Mr. W. Chilton and Mr. J. M. Newton, B.Sc., of the Brush Electrical Engineering Company, a special meed of thanks is due for the results of some of their experiments on blading.

The general scheme of this volume was decided on after much consideration, and the author finally adopted the plan of giving, without prior proof, important rules and formulas in a shape convenient for immediate practical application. The demonstrations are proceeded with later, in the belief that they will be the more readily followed by the average reader, when he has previously been impressed with the utility of the result.

The opening Chapters are accordingly devoted to an explanation of the nature of a turbine, and to the setting forth of the elementary theory of guide blades and moving buckets. The uses of the Mollier diagram are described here in detail, but, in accordance with the policy already explained, the theory of the construction of this diagram is deferred to a later Chapter. An explanation of the terms "efficiency ratio" and "reheat factor" follows, and the uses of correction curves are next explained. For the excellent and comprehensive set reproduced in Figs. 37 to 42

the author is indebted to Mr. K. Baumann, of the British Westinghouse Company.

The way having been thus cleared, a process is described by which the general dimensions, of a turbine of any type for any specified output and steam consumption, can be got out very simply and easily. This forms the subject of Chapter VII., and has been developed from an article by the author which appeared in ENGINEERING, vol. lxxxiv., page 799.

Methods of analysing in detail a provisional design, thus arrived at, form the subject of succeeding Chapters. Of the two Chapters in which the equations for the flow of steam through groups of blades are established, Chapter XIII. is wholly new, but the solution of the more difficult problem dealt with in the next Chapter was originally published in ENGINEERING.

The theory of the reaction-steam turbine constitutes in reality a problem in the calculus of finite differences, and the mathematical work, in the two Chapters in question and in Chapter XVIII., might have been abbreviated somewhat by assuming certain of the established formulas of this branch of mathematics. Since, however, this subject is relatively little known to engineers, it was thought preferable to assume no knowledge of it, and to derive the formulas in question directly from elementary considerations.

Any satisfactory theory of a steam turbine must be self consistent. It cannot be considered legitimate, at least from the scientific standpoint, to use one set of constants to estimate the efficiency of the turbine, and an independent set to predict its output. So long as this is necessary, the theory must be held as incomplete and unsatisfactory. This is still the case with the theory of the velocity-compounded wheel, and there have been accordingly numerous failures, to realise guarantees, when this type of turbine has been taken up by builders having had no practical experience of its peculiarities. It is to be hoped that, as time goes on, the test data now pigeon-holed in the archives of manufacturers of this type will become available for detailed scientific analysis.

Mr. B. A. Raworth has been good enough to undertake the heavy task of perusing the whole of the proof sheets, and to him the author is deeply indebted for many most valuable suggestions and emendations.

H. M. M.

26, ADDISCOMBE ROAD,
CROYDON.
November, 1912.

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ADDENDUM TO CHAPTER VII.

THE curve plotted in Fig. 43 can also be used for getting out the approximate proportions of an exhaust-steam turbine. In fact, the latter may be considered as being simply one-half of a high-pressure turbine. Thus an exhaust turbine with a coefficient of 60,000 may be regarded as equivalent to a high-pressure turbine with a coefficient of 120,000. From Fig. 43 the latter should have an efficiency ratio of about 70 per cent., and the corresponding efficiency ratio of the equivalent exhaust turbine will therefore be about $0.70 \times \frac{1 + R}{2 R}$, where R denotes the reheat factor for the high-pressure turbine. Taking this, from the table on page 43, as 1.06, the brake efficiency ratio of the proposed exhaust turbine becomes 68 per cent., which is about the figure which might be expected from a really large low-pressure turbine with a coefficient of 60,000.

Exhaust-steam turbines do not however, in general, run to large sizes, and as shown by the point marked R, in Fig. 43, there may be with small turbines a very marked falling off in the efficiency realised with a given coefficient. For a 600-kilowatt single-flow exhaust turbine the size factor may be taken as 0.96.

The author is indebted to the builders of, perhaps, the best of the continental makes of impulse-steam turbines for data, which would enable an additional point to be plotted on the curve, Fig. 43. The machine concerned, which is of their latest pattern, is of 3000 kilowatt capacity, running at 1545 revolutions per minute, and comprises 15 stages. The value of λ is 130,000, and the brake efficiency ratio, reduced to the conditions of Fig. 43, is 71.5 per cent. This it will be found is in excellent agreement with the curve as plotted.

ERRATA.

Page 18, line 9, *for* "0.82" *read* "0.87."

„ 28, line 7, *for* "D. Napier" *read* "R. D. Napier."

„ 35, line 5, *for* "lead squares" *read* "least squares."

„ 59, line 9 from bottom,

for "$6 \times \left(\frac{d}{10}\right)^2 \times \left[\frac{\text{R. P. M.}}{100}\right]$" *read* "$6 \times \left[\frac{d}{10}\right]^2 \times \left[\frac{\text{R. P. M.}}{100}\right]^2$."

„ 69, line 14, *for* "$A = \frac{\rho}{x} = \left(\frac{1}{x}\right)^{\gamma - 1}$" *read* "$A = \frac{\rho}{x} = \left(\frac{1}{\rho}\right)^{\gamma - 1}$."

„ 93, line 6, *for* "$A_N = \frac{V_N}{V_0}, \frac{p_N}{p_0}$" *read* "$A_N = \frac{V_N}{V_0} \times \frac{p_N}{p_0}$."

„ 137, line 14 from bottom, *for* "$2\,\omega \sum_{n=1}^{n=\frac{N}{2}} r^3{}_n$" *read* "$2\,\omega \sum_{n=1}^{n=\frac{N}{2}} r_2{}^2{}_n$."

„ 156, line 4 from bottom, *for* "2.236" *read* "2.356."

„ 162, column 4, *for* "Bending moment in pounds" *read* "Bending moment, inch-pounds."

„ 170, line 7, *for* "$w = 68\,\Omega\sqrt{\frac{p_1\left(1 - \frac{1}{x^2}\right)}{V_1 N + \log_e x}}$,"

read "$w = 68\,\Omega\sqrt{\frac{p_1\left(1 - \frac{1}{x^2}\right)}{V_1 (N + \log_e x)}}$."

„ 289, line 5 from bottom, *for* "2000" *read* "3000."

STEAM TURBINES:

THEIR THEORY AND CONSTRUCTION.

CHAPTER I.

PRELIMINARY IDEAS.

IN all practical forms of turbine, whether steam or hydraulic, the working fluid is directed in a jet or jets from one or more nozzles or guide blades on to a set of vanes or buckets mounted on the rim of a wheel, which is thereby impelled round, or, alternatively, the working fluid is passed under pressure into the moving element of the turbine, and, issuing thence from nozzles set approximately tangentially, drives the wheel by its reaction.

IMPULSE TURBINES.

The simplest and most easily understood form of turbine is the Pelton wheel, the characteristic features of which are clearly set forth in Fig. 1 (page 2). Here a jet of fluid, issuing under pressure from a suitable orifice, is directed into a set of cup-formed buckets mounted on a wheel. The Pelton wheel constitutes, as stated, the simplest form of impulse turbine, and exhibits more clearly perhaps than other varieties the essential characteristics of the type.

In the supply pipe the speed of flow is low, but the pressure is high. On the other hand, as the jet leaves the nozzle its velocity, and consequently its kinetic energy, is very great. This increase of kinetic energy must have been derived from some source, and is, in fact, obtained at the expense of the potential energy represented by the pressure of the fluid in the supply pipe. The nozzle thus converts pressure into velocity, or potential energy into kinetic.

The water, which in the supply pipe had a pressure of several atmospheres, issues at atmospheric pressure. That this is so is

shown very clearly by Fig. 2, which is reproduced from a photograph of a jet issuing from a well-formed nozzle. Were the pressure in the interior of the jet higher than that of the atmosphere, the jet would swell instead of remaining practically parallel as shown.

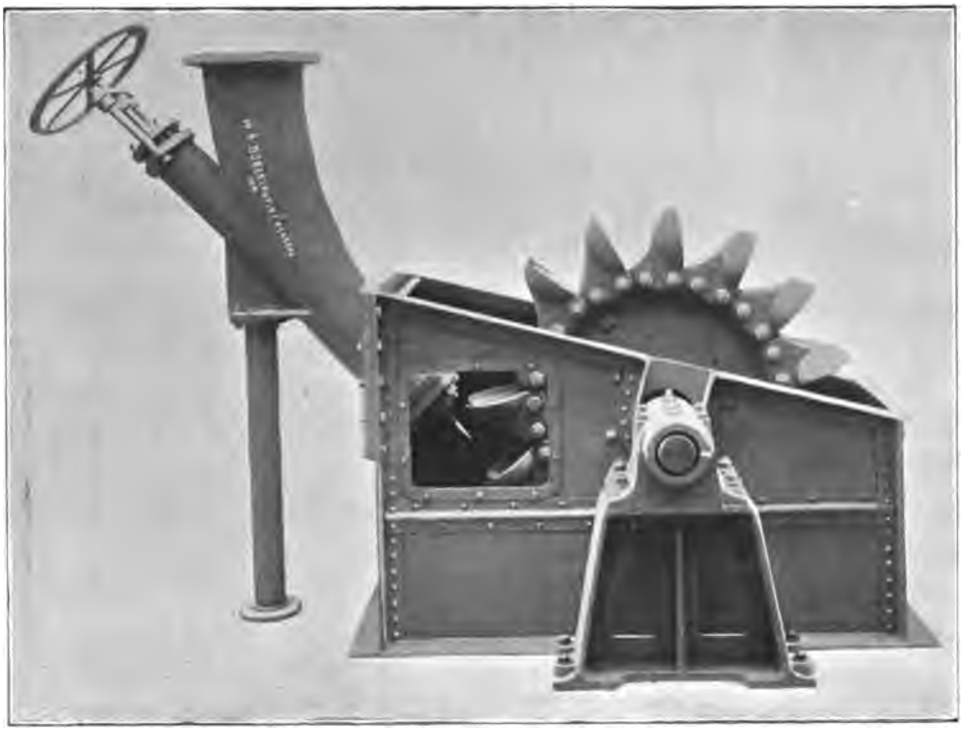


Fig. 1. Pelton Wheel.

After passing through the buckets of the wheel the water falls clear and passes off to the tail race. In the latter it is also under atmospheric pressure, and hence its pressure has not been changed by its passage through the moving buckets of the wheel. This is the essential characteristic of the impulse type of turbine. The fall in pressure occurs solely within fixed nozzles or guide blades.

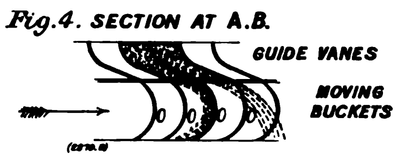
On reference to Fig. 1, it will be obvious that several jets might be arranged to play upon the same wheel, and that it is purely a matter of convenience, and alters nothing in the essentials of the system, whether this arrangement is adopted, or whether the working fluid is supplied through a single nozzle. If there are sufficient nozzles to render the supply continuous around the whole circumference of the

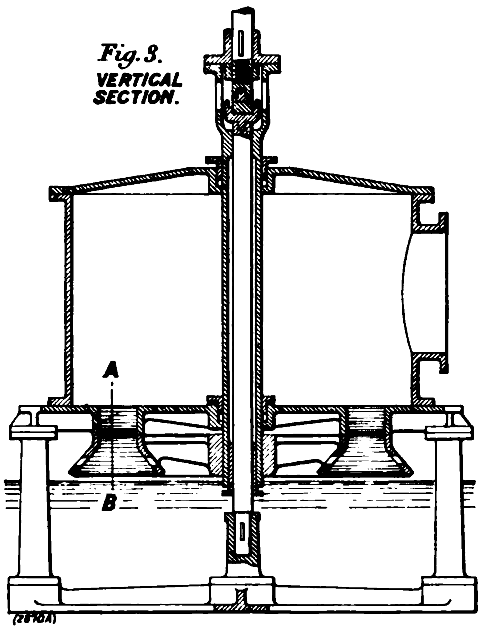
wheel, the latter is said to work with complete admission. In all other cases the admission is partial.

In the case of a Pelton wheel the nozzles all lie in the plane of the wheel, and some steam turbines have been successfully constructed to operate in the same way. More generally though, impulse steam turbines are derived from another pattern of impulse wheel known as the Girard turbine.

The general characteristics of a Girard water turbine are shown in Fig. 3. In this case the nozzles or guide vanes are arranged above

Fig. 2. Flow from a Nozzle.





Figs. 3 and 4. Girard Water Turbine.

the moving wheel, on to which they deliver jets inclined at an angle of 15 deg. to 20 deg. to its plane, as indicated by a section through A B (Fig. 4). The buckets have the simple curved form shown, and the water, it will be seen, does not fill them. As in the case of the Pelton wheel, the water leaves the nozzles under atmospheric pressure, and is not altered in pressure by passing through the buckets. Girard turbines can be constructed to work either with complete or with partial admission. That is to say, there may be a complete ring of guide blades, or only a few, subtending a mere fraction of the whole circumference of the wheel. The De Laval turbine is an example of a steam Girard wheel.

The possibility of working with partial admission constitutes the chief practical advantage of the impulse system of working. As it is unnecessary that the fluid shall be supplied all round the wheel, the size of the latter is independent of the quantity to be passed. Thus, with a given nozzle and given pressure in the supply pipe, the same results will be obtained if we use a Pelton wheel 6 ft. in diameter running at 100 revolutions per minute, or one 3 ft. in diameter running at 200 revolutions per minute.

REACTION TURBINES.

The characteristic feature of reaction turbines is that the fluid actuating them changes in pressure as it passes through the moving element of the turbine.

Perhaps the simplest form of reaction turbine is that shown in Fig. 5, which represents an experimental form tried by Sir Charles A. Parsons in 1893. It consists of an arm mounted on a hollow shaft and counterbalanced as indicated. The arm is hollow and terminates in a nozzle directed tangentially as shown. Steam admitted through the hollow shaft expands through this nozzle down to the pressure in the chamber. A high velocity of efflux is generated by this expansion, and the arm is driven round by the reaction of the jet.

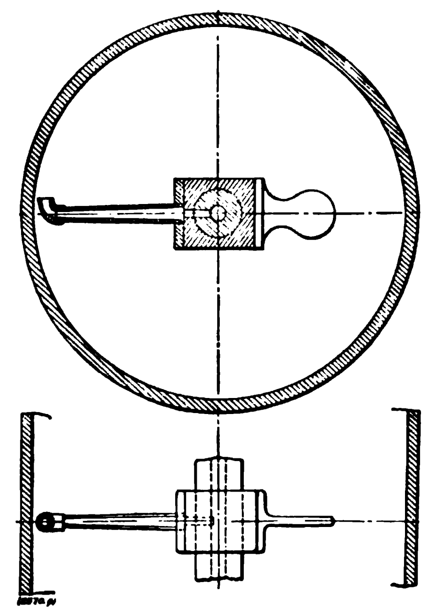


Fig. 5. Simple Reaction Turbine.

Such a turbine is known as a pure reaction turbine. Ordinary reaction turbines are, however, constructed on somewhat different lines, and may be considered as Girard wheels, having complete admission, but modified so that there is a fall of pressure in the moving buckets as well as in the guide blades.

The moving buckets have accordingly the form indicated in Fig. 6, and have the same width radially as the guide blades, instead of being flared out as shown at B in Fig. 3. With the form of

bucket shown in Fig. 6, the area available for flow is less at the discharge from the bucket than it is at the entrance, hence the fluid must increase in velocity in passing through the wheel, since the quantity leaving is the same as that entering, and it has to leave through a narrower opening. To get this increase of velocity there must be a fall of pressure within the moving bucket, and thus the pressure in the clearance p_2 is intermediate between p_1, the pressure above the nozzles, and p_3 the final pressure of discharge. To prevent excessive loss by leakage, owing to this pressure in the clearance space, the whole wheel has to be closely boxed in and run with fine clearances.

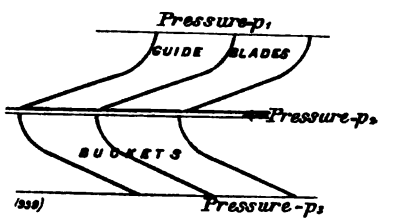


Fig. 6.
Blades of Typical Reaction Turbine.

Turbines in which the fluid passes through the machine from one side of the wheel to the other are known as parallel-flow turbines. Turbines of the Pelton type, on the other hand, are called tangential-flow wheels, because the direction of the jet is a tangent to a circle passing through the centres of the buckets. There is, in addition, another type known as radial-flow machines. In these the direction of flow is similar to that of the fluid in a centrifugal pump. All three types, viz., parallel-flow, tangential-flow and radial-flow turbines, may be built to work either on the impulse or on the reaction system; the distinction between which, rests solely on the fact that in the reaction turbine a conversion of pressure into velocity takes place within the moving element of the machine, whilst in the case of an impulse turbine this conversion occurs wholly within fixed elements of the turbine.

CHAPTER II.

GUIDE BLADES, NOZZLES, AND THE MOLLIER DIAGRAM.

FROM the preceding Chapter it will be apparent that a guide blade or nozzle forms an essential element of every turbine. Within these blades or nozzles there is a conversion of pressure head into velocity head, and the velocity thus generated can be calculated when the change of pressure is known.

In the case of water or other non-elastic fluid, the velocity, in feet per second, generated in a frictionless nozzle is given at once by the well-known equation

$$\frac{v^2}{2g} = h,$$

where h denotes the pressure drop expressed in feet of head. An expression exactly similar in form can be used to give the theoretical velocity of outflow when the issuing fluid is steam.

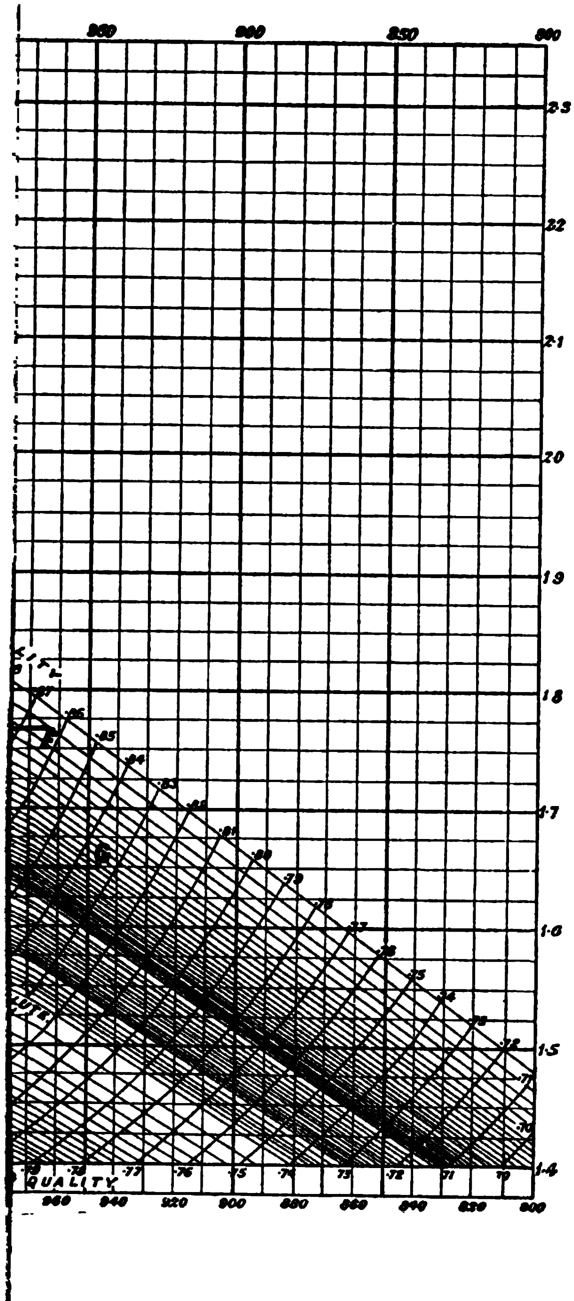
We may then write—

$$v = 224\sqrt{u} \qquad . \quad . \quad . \quad . \quad . \quad . \qquad (1)$$

Here the quantity u is expressed in British Thermal Units, and is known as the "available heat," as the "thermodynamic head," or as the "heat drop." The derivation of this formula will be given later, but for the present it may merely be stated that this quantity u may be read direct from what is known as the Mollier diagram, of which an example is reproduced in Fig. 7. For the method of constructing this diagram, reference must be made to Chapter XVI. on Thermodynamic Principles, but a knowledge of these is by no means essential to its intelligent use.

Suppose, for example, that it is required to find the value of u in the case of a nozzle discharging into a space in which the pressure is, say, 14 lb. absolute, and that this nozzle is supplied with steam at 180 lb. absolute, which we may assume to be superheated till its temperature is 420 deg. Fahr. On the diagram will be found a number of curved lines marked "Constant Pressure Lines." These are crossed on the left-hand side of the diagram by a number

AGRAM.



To F

of lines marked with different temperatures, and on the right-hand side with a similar set of lines marked "Constant Quality Curves." On the diagram find the point A, where the curve of 180 lb. pressure cuts the line corresponding to a temperature of 420 deg. Fahr.

On referring to the scale of Total British Thermal Units, at the foot of the diagram, it will be seen that 1 lb. of steam under the conditions stated, has a total heat content of about 1223.6 B.Th.U. Suppose then that this steam is expanded through a frictionless nozzle down to a pressure of 14 lb. absolute. Then the pound of steam as it leaves the nozzle has acquired velocity, and consequently possesses a certain kinetic energy. This energy can only have been obtained by the conversion, into this form of energy, of some of the 1223.6 units of total heat the steam originally possessed. If we call the quantity thus converted u, the value of u is found by drawing the line AB to cut at B the line of 14 lb. pressure. From the scale at the bottom it will be seen that steam in the condition represented by B contains 1032.8 B.Th.U., or 190.8 units less than before it expanded through the nozzle. These units have, in fact, been converted into kinetic energy. being the value of u, which is to be substituted in equation (1). If v be the velocity of 1 lb. of steam on leaving the nozzle, its kinetic energy will be $\frac{v^2}{2g}$ ft.-lb., whilst the u units of heat are equal to $u \times 778$ ft.-lb. Equating these two quantities we get

$$v^2 = 64.4 \times 778\, u,$$

or

$$v = 224 \sqrt{u}.$$

Taking the value found for u—viz., 190.8 B.Th.U. the theoretical velocity of efflux will be

$$v = 224 \sqrt{190.8} = 3094 \text{ ft. per second,}$$

which is equal to the speed of the fastest projectiles.

It should be noted, in passing, that with a simple convergent nozzle passing non-superheated steam, the speed of efflux never exceeds some 1500 ft. per second or so, and that to obtain a velocity of the above order it would be necessary to use a divergent nozzle. The explanation of this anomaly will be given later.

Nozzles are never quite free from friction, so that the actual velocity generated is always less than the theoretical. Thus the actual kinetic energy of the pound of steam on issue might in

practice be, say, 10 per cent. less than the theoretical, and equal, therefore, to 171.7 heat units in place of 190.8. Hence the total heat content of the pound of steam on discharge would then be 1223.6 – 171.9 = 1051.9 B.Th.U. in place of 1032.8 B.Th.U., as was the case when there were no frictional losses. The actual condition of the steam on discharge can then be found by setting off, in Fig. 7, the point D on the lower scale at 1051.9 B.Th.U., and drawing the vertical line D C to cut at C the curve corresponding to 14 lb. pressure.

The actual velocity ot flow from the nozzle will in this case be equal to

$$224 \sqrt{171.7} = 2934 \text{ ft. per second.}$$

It will be seen from the diagram that the point B lies between the "quality curves" 0.88 and 0.89, so that the quality of the steam corresponding to a frictionless flow is about 0.883. Referring to a steam table, it will be found that the specific volume of dry steam at 14 lb. absolute is 28.02 cub. ft. per pound. Hence the actual volume of 1 lb. on discharge will be 28.02 × 0.883 = 24.7 cub. ft. per lb. In other words, a part of the steam has condensed as it expanded, and each pound as finally delivered from the nozzle contains 0.116 lb. in the form of moisture, and 0.884 lb. in the form of vapour.

The quality, or as it is often called, the "dryness fraction," corresponding to the point C is, it will be seen, about 0.904, so that nozzle friction dries the steam. Thus its volume as delivered in this case is 28.02 × 0.904 = 25.4 cub. ft. per lb.

The area needed at the point of efflux is given by the relation

$$\Omega \times v = w \times V,$$

where Ω denotes the required area in square feet, v the actual velocity of flow, w the weight passed per second, and V the actual volume of 1 lb. of steam at discharge. In the first case, therefore, the area needed per pound of steam passed per second is

$$\Omega = \frac{V}{v} = \frac{24.7}{3094} = 0.00800 \text{ sq. ft.} = 1.150 \text{ sq. in.}$$

In the second case, where there is a 10 per cent. loss of energy by friction, the corresponding figure is

$$\frac{25.4}{2934} = 0.00868 \text{ sq. ft.} = 1.250 \text{ sq. in.}$$

In the case taken above the steam was initially superheated,

but the method of using the diagram is the same if the steam be wet. Thus, suppose steam at 10 lb. pressure and 99 per cent. dry is to be expanded through a nozzle down to 0.6 lb. absolute. In this case the initial condition of the steam is represented by the point E, and its final condition, if there be no friction loss, by the point F. The length of the line E F is about 168 heat units, and the theoretical speed of efflux is $224\sqrt{168} = 2903$ ft. per second. The steam on delivery will have a dryness fraction of about 0.862. Corrections for nozzle friction can be made exactly as before.

A nozzle, it will be seen, is a device by which part of the heat of the steam is converted into kinetic energy, and the latter is finally converted by the action of the moving buckets of the turbine into mechanical work on the turbine shaft. If both nozzles and buckets had perfect efficiency, the whole of the u units of heat which are available for producing flow through the nozzle would appear as useful work on the turbine shaft.

The noteworthy point is that out of the total heat of the steam, which in the case taken first was 1223.6 B.Th.U., only 190.8 units are thus capable of conversion into work, even if there were no frictional losses. Of the original total heat, only this small fraction is even theoretically "available" for conversion into useful work. It is for this reason that the quantity u is known as the "available heat." A further discussion of this point will be found in Chapter XVI., but in all cases the quantity u can be measured off the Mollier diagram in the manner above set forth.

Flow Under Small Differences of Pressure.

When the pressure drop through the guide blades is small, the scale of the Mollier diagram is too contracted to permit of a reasonably accurate measurement of u. The theoretical velocity of efflux is then given by the relation

$$v = 8.02\sqrt{144\,V\,\Delta p} \text{ ft. per second} \quad . \quad . \quad . \quad (2)$$

or

$$v = 96.2\sqrt{V\,\Delta p}.$$

Here Δp denotes the small pressure drop in pounds per square inch, and V may then be taken either as the specific volume of the steam before the expansion, or as its volume after discharge, the two, in the case of small drops of pressure differing but little. If V is

taken as the mean of these two values, the expression in question can be used with fair accuracy for drops of pressure which are not very small. As before, owing to frictional losses, the actual velocity of outflow will be less than given by this expression.

From this formula it follows that in the absence of friction the weight discharged from a nozzle, of suitable form, under a small pressure drop is given by the relation

$$w = 63 \, . \, A \sqrt{\frac{\Delta p}{V}} = 80 \, D^2 \sqrt{\frac{\Delta p}{V}}$$

where w denotes the weight discharged in pounds per second and A the area of the orifice in square inches, whilst D denotes the diameter in inches of the outlet of the nozzle if round. The use of round nozzles is, however, undesirable in turbine practice, as experiment shows that they lead to inefficiency.

In using the above expression, it should be noted that Δp is to be reckoned small only when it is a small fraction of the higher pressure. Thus if the initial pressure is 100 lb. absolute, a value of $\Delta p = 1$ lb. per sq. in. is to be reckoned small; but if the original pressure were 2 lb. per sq. in. absolute, a value of Δp equal to 1 lb. would be reckoned large, since it is a large fraction of the initial pressure.

Formula (2) may be established by considering an ideal indicator diagram, such as is represented in Fig. 8. Suppose this refers to a cylinder containing exactly 1 lb. of steam. Let the initial pressure measured in pounds per square inch be denoted by A F, and the back pressure by A E. Then, if this pound of steam were used in a steam-engine cylinder, the effective work done would be represented by the area F B D E.

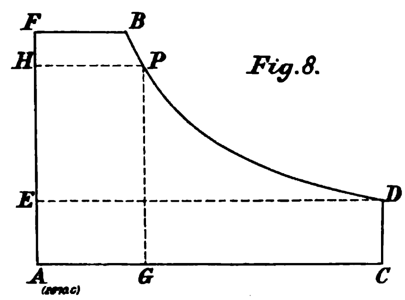


Fig. 8.

If in place of expanding in a cylinder the steam expands through a frictionless nozzle, it still does the same amount of work; but this work is now represented by the kinetic energy of the jet.

Hence, if the area F B D E represents the work, expressed in foot-pounds, which could be done by 1 lb. of steam in a cylinder, its velocity in flowing through a frictionless nozzle when the pressure

on the inlet side is A F, and on the outlet side A E, is given by the relation

$$\frac{v^2}{2g} = \text{F B D E}.$$

Now what is true for the diagram as a whole is true also for its constituent parts. In place of the back pressure being equal to A E, suppose it is equal to P G, then the work done by the 1 lb. of steam in the cylinder will be very nearly equal to

$$144 \,.\, \text{F H} \times \frac{\text{F B} + \text{H P}}{2} \text{ ft.-lb.}$$

Putting

$$\text{F H} = \Delta p \text{ and } \frac{\text{F B} + \text{H P}}{2} = \text{V},$$

we get

$$\frac{v^2}{2g} = 144 \,.\, \Delta p \,.\, \text{V},$$

which, on reduction, gives equation (2), *supra*.

It was shown above that in every case the velocity of flow is given by the relation

$$\frac{v^2}{2g} = \text{the area F B D E}.$$

In a frictionless nozzle the steam must expand without addition or subtraction of heat. It moves so fast that there is no time for it to lose heat to, or pick up heat from, the nozzle walls by conduction, and thus, in the absence of friction, the expansion will be adiabatic. Hence the expansion line BD in Fig. 8 is an adiabatic curve.

In the case of non-superheated steam the equation to the curve of adiabatic expansion is approximately given by the relation

$$p\,\text{V}^{\frac{10}{9}} = p_0\,\text{V}_0^{\frac{10}{9}},$$

where p_0 denotes the initial absolute pressure of the steam, and V_0 the initial volume occupied at this pressure by 1 lb.

Actually the index varies a little, both with the initial pressure of the steam and with its quality. If not quite dry initially, the index is a little less than it otherwise would be. The variations are, however, small, and of no particular moment in practical applications.

For steam initially dry Zeuner gives 1.135 as the best average value of the index, which, it will be seen, is a little greater than $\frac{10}{9}$.

In the ordinary treatises on the steam engine it is shown that

if the law of expansion in a steam cylinder is $p\,V^{\gamma}$ = constant, the total work done will be:

$$\text{Work done} = \frac{144\,.\,\gamma}{\gamma - 1}\,.\,(p_0\,V_0 - p_1\,V_1),$$

where p_0 denotes the initial pressure in pounds per square inch, V_0 the initial volume in cubic feet, whilst p_1 denotes the terminal pressure, and V_1 the corresponding volume.

Hence the area of F B D E, Fig. 8, is equal to

$$\frac{144\,\gamma}{\gamma - 1}\,.\,(p_0\,V_0 - p_1\,V_1),$$

if

$$p_1 = \frac{p_0}{x} \text{ and if } V_1 = \rho\,V_0;\ \rho \text{ being the ratio of expansion,}$$

this can also be written as work done

$$= \frac{144\,\gamma}{\gamma - 1}\,.\,p_0\,V_0\left(1 - \frac{\rho}{x}\right). \quad . \quad . \quad . \quad . \quad . \qquad (3)$$

Moreover, since

$$p\,V^{\gamma} = p_0\,V_0^{\gamma},$$

we get

$$\frac{p_0}{x}(\rho\,V_0)^{\gamma} = p_0\,.\,V_0^{\gamma}.$$

Whence (3) may also be written in the forms

$$\text{Work done} = \frac{144\ \gamma}{\gamma - 1}\,.\,p_0\,V_0\left(1 - \rho^{1-\gamma}\right). \quad . \quad . \quad . \qquad (4)$$

$$= \frac{144\,.\,\gamma}{\gamma - 1}\,.\,p_0\,V_0\left(1 - \left(\frac{1}{x}\right)^{\frac{\gamma - 1}{\gamma}}\right). \quad . \quad . \quad . \quad . \qquad (5)$$

As already stated, when the expansion occurs through a frictionless nozzle the work due is represented by the kinetic energy of the jet. Hence, taking as unit 1 lb. of steam at an initial pressure p_0 lb. per square inch, and an initial volume of V_0 cubic feet per pound, then if the final pressure is p_1 and the final volume V_1, we have

$$\frac{v^2}{2g} = \frac{\frac{10}{9} \times 144}{\frac{10}{9} - 1}(p_0\,V_0 - p_1\,V_1)$$

$$= 1440\,(p_0\,V_0 - p_1\,V_1)$$

$$= 1440\,p_0\,V_0\left(1 - \frac{\rho}{x}\right). \quad . \quad . \quad . \quad . \quad . \qquad (6)$$

$$= 1440\,p_0\,V_0\left\{1 - \left(\frac{1}{\rho}\right)^{\frac{1}{9}}\right\}. \quad . \quad . \quad . \quad . \qquad (7)$$

$$= 1440\,p_0\,V_0\left\{1 - \left(\frac{1}{x}\right)^{\frac{1}{10}}\right\}. \quad . \quad . \quad . \qquad (8)$$

For superheated steam the index is about 1.30 instead of $\frac{10}{9}$. and hence the law of expansion changes if the steam loses its superheat in the process of its expansion.

Such cases are best dealt with by means of the Mollier diagram, or by corresponding arithmetical methods explained in Chapter XVI., on Thermodynamic Principles.

A very close estimate of the velocity can, however, be obtained by a formula of the type of (4) and (5); if a suitable value of γ is chosen.

Thus, in the case already taken (page 6) of a nozzle supplied with steam at a pressure of 180 lb. absolute, and superheated to 420 deg. Fahr., the specific volume V_0 of 1 lb. is about 2.731 cub. ft per lb. With $p_1 = 14$ it was found above that $V_1 = 24.70$ cub. ft. per lb.

To find the appropriate value of γ we have the relation

$$p_0 V_0^{\gamma} = p_1 V_1^{\gamma},$$

or

$$\log p_0 + \gamma \log V_0 = \log p_1 + \gamma \log V_1$$

whence

$$\gamma = \frac{\log p_0 - \log p_1}{\log V_1 - \log V_0} = \frac{\log x}{\log \rho}.$$

Now

$$x = \frac{180}{14} \text{ and } \rho = \frac{24.7}{2.731}.$$

Hence

$$\log x = 1.10914 \quad ; \quad \log \rho = 0.93104,$$

and

$$\gamma = \frac{\log x}{\log \rho} = \frac{1.10914}{0.93104} = 1.1598.$$

Hence from equation (4) we have

$$\text{Work done} = \frac{v^2}{2g} = \frac{144 \times 1.1598}{0.1598} \,.\, 180 \times 2.731 \left\{1 - \frac{24.7}{2.731} \cdot \frac{14}{180}\right\} = 152{,}500 \text{ ft.-lb.}$$

$$\therefore v = 8.02 \,.\, \sqrt{152{,}500} = 3132 \text{ ft. per second.}$$

The value found from the Mollier diagram was 3094. ft. per second, or practically the same.

Critical Velocity of Flow.

As already mentioned on page 7 *ante*, the maximum velocity with which steam (not superheated) can flow from a converging nozzle does not exceed some 1500 ft. or so. This anomaly was discovered by Mr. R. Napier, in the course of experiments on the

discharge of steam from orifices carried out in 1866-67. He found that starting with an internal pressure of, say 100 lb., the weight discharged per second increased as the external pressure was lowered, until the latter attained a certain critical value, which he found was then somewhere about half the internal absolute pressure. Any further reduction of the external pressure gave no increase in the quantity discharged. He concluded that if the absolute external pressure was less than one-half the internal, that the expansion of the steam was not completed until some time after it had cleared the orifice, and that with convergent nozzles the pressure at the plane of discharge in no circumstances became less than half the internal pressure. He also found that by adding a diverging mouthpiece, or addendum, to a convergent nozzle, the expansion of the steam could be completed within this nozzle, and the theoretical velocity of discharge approximately attained. Such convergent-divergent nozzles were introduced into turbine practice by Dr. de Laval, of Stockholm, and should be adopted whenever it is desired to utilise a heat drop of much more than, say, 45 B.Th.U. in the case of non-superheated steam, or of some 63 B.Th.U. in the case of highly superheated steam.

The theory of the anomaly was given by Rankine in 1868.

The weight w (expressed in pounds per second) discharged from an orifice Ω square feet in area is given by the relation

$$w = \frac{v\,\Omega}{V} \qquad (9)$$

where v denotes the velocity of outflow and V the specific volume of the steam on its discharge. If, then, we start with dry steam at 100 lb. pressure and calculate the velocity of discharge from equation (8) *ante*, when the external pressure is made successively 80 lb., 60 lb., 50 lb., &c., per sq. in. absolute, we get the figures given in column 2 of Table I. We can also calculate V, the specific volume on discharge from the relation

$$p\,V^{\frac{10}{9}} = p_0\,V_0^{\frac{10}{9}}$$

and we thus get the value of V given in the third column of the table. If Ω is made equal to 1 sq. ft., the weight discharged per second is from (9), numerically equal to $\frac{v}{V}$, and this is given in the last column of the table.

TABLE I.

External Pressure. Pound per Square Inch (Absolute). p	Calculated Velocity of Flow. v Feet per Second.	Volume of 1 lb. of the Steam at Discharge. V Cubic Feet.	Weight Discharged per Second. $\frac{v}{V_2}$
100	0.	4.33	0
80	937.6	5.294	178
60	1409.0	6.860	205
50	1639.0	8.081	202
40	1874.0	9.879	190
30	2440.0	18.44	131

According to this calculation the discharge attains a maximum for a certain value of p, lying between 0.5 p_0 and 0.6 p_0 and afterwards diminishes. Actually no diminution takes place, but when the pressure attains a certain critical value at the outlet, it remains constant at that value, and however much the external pressure be subsequently lowered, the remainder of the expansion of the issuing steam is completed after it is clear of the orifice. The physical explanation of this peculiarity was given by Osborne Reynolds, whose reasoning may be summarised as follows:

The velocity of sound is that with which any impulse is transmitted through an elastic system. The speed of sound in steel (for example) is about 18,000 ft. per second, so that if one end of steel wire 100 ft. long were suddenly pulled, no indication of this would reach the other end till $\frac{1}{180}$th of a second later.

Now suppose a jet of steam issuing from an orifice with a velocity v, and let S be the speed of sound in the steam. If the external pressure is suddenly lowered, "news" of this operation will be transmitted back along the jet with a velocity equal to S–v. If, however, the ratio of the external to the internal pressure is such that v is equal to S, then the "news" of any further lowering of the outer pressure cannot be telegraphed back to the interior of the reservoir. The flow of the fluid to the orifice is then quite unaffected, and the discharge accordingly becomes constant. The value of this critical pressure ratio was determined by Rankine as follows:—

The weight discharged per second is by equation (9)

$$w = \frac{v\,\Omega}{V}.$$

Hence

$$w^2 = \frac{v^2\,\Omega^2}{V^2}.$$

The value of v^2 by equation (8) is given by

$$\frac{v^2}{2g} = 1440 \,.\, p_0\,V_0\left\{1 - \left(\frac{1}{x}\right)^{\frac{1}{10}}\right\}.$$

Also

$$V = \rho\,V_0 \,. = x^{\frac{9}{10}} \,.\, V_0.$$

We thus get

$$w^2 = \frac{v^2\,\Omega^2}{V^2} \,. = 2g \,.\, 1440 \,.\, \frac{p_0\,V_0}{x^{\frac{9}{5}}\,V_0^2}\left(1 - \left(\frac{1}{x}\right)^{\frac{1}{10}}\right)$$

$$= 2g \,.\, \times\, 1440 \,.\, \frac{p_0}{V_0}\left[\left(\frac{1}{x}\right)^{\frac{9}{5}} - \left(\frac{1}{x}\right)^{\frac{19}{10}}\right].$$

Hence w^2 will be a maximum when $\left(\frac{1}{x}\right)^{\frac{9}{5}} - \left(\frac{1}{x}\right)^{\frac{19}{10}}$ is a maximum.

Putting $Z = \frac{1}{x}$. we get $Z^{\frac{9}{5}} - Z^{\frac{19}{10}}$ = a maximum.

Differentiating and equating to zero, we have

$$\frac{9}{5}Z^{\frac{14}{5}} - \frac{19}{10}Z^{\frac{29}{10}} = 0.$$

$$\therefore \quad 18\,Z = 19\,Z^{\frac{1}{10}}$$

Therefore,

$$\log 18 + \log Z = \log 19 + \frac{1}{10}\log Z.$$

$$\therefore \quad \log Z = -\frac{10}{9}.\log\frac{19}{18}$$

$$= -\quad 0.2348$$

$$= \bar{1}.7652.$$

Whence

$$Z = \frac{1}{x} = 0.582.$$

If the value of the index is taken as 1.135, in place of $\frac{10}{9}$, we get

$$Z = 0.577 \text{ and } x = 1.73.$$

From Napier's experiments Rankine deduced the following simple empirical formula for the weight discharged per second when the critical value of x is exceeded :—

$$w = \frac{A \,.\, p}{70},$$

where A denotes the area of the orifice in square inches, and p the absolute pressure, in pounds per square inch, of the steam on the inlet side of the nozzle.

Another empirical formula has been given by Professor Rateau. This is a little more complicated than Rankine's, but is more precise, particularly when p is small. Rateau's formula may be written as

$$w = \frac{[16.57 - \log p]\, p \,.\, \mathrm{A}}{1000}.$$

It was at one time assumed, as the result of Osborne Reynolds' physical explanation of the critical pressure ratio, that the speed of any gas escaping from an orifice could never exceed the speed of sound. As already stated, however, Napier found that by adding a divergent cone to his convergent nozzle, much higher velocities could be reached; approximating to those given by equation (1), even when u is very large. In such nozzles the steam expands down to a pressure 0.577 p_0 at the throat, and then completes its expansion in the divergent extension. The weight discharged per second from such a nozzle may therefore be found by using either Rankine's or Rateau's formula, and putting for A the area of the throat in square inches. The area needed at the outlet end of the divergent cone is then determined as explained on page 8, *ante*. It is, however, found desirable in practice to make the area of the outlet a little less than that thus calculated, but this point will be developed further in another Chapter.

CHAPTER III.

NOZZLE AND GUIDE-BLADE EFFICIENCY.

IN the preceding Chapter rules have been given for calculating the theoretical velocity of efflux from nozzles, and also the theoretical discharge in pounds per second. With actual nozzles there are frictional losses which diminish the velocity; and in some cases the discharge is further affected by the fact that the jet does not entirely fill the opening, but forms a *vena contracta*. Thus the actual discharge of steam from a sharp-edged round hole in a thin plate is only 0.62 of the theoretical for moderate drops of pressure, rising to 0.82 of the theoretical, when the critical velocity (as defined in the previous Chapter) is attained.

If v be the theoretical velocity of outflow, the actual velocity is equal to $c\,v$, where c is known as the coefficient of velocity. Similarly if Ω be the actual area in square feet, the effective area is $C\,\Omega$, where C is known as the coefficient of contraction. The volume discharged per second is therefore $c \,.\, C \,.\, v \,.\, \Omega$, instead of $v\,\Omega$. Turbine guide blades and nozzles are generally of such a form that C is practically unity, and the volume discharged per second is then equal to $c \,.\, v\,\Omega$. The weight discharged, per second, is obtained by dividing this quantity by the specific volume of the steam.

Many experiments have been made on the efficiency of nozzles, but only a few of these have given reliable results. In some cases, for instance, no attempt was made to determine the initial state of the steam. In one set of somewhat elaborate experiments, for example, there is reason to believe that the steam, as it reached the nozzle, contained 10 per cent. to 13 per cent. of moisture. The dryness was not, however, actually measured, and hence it is impossible to draw any trustworthy conclusions from the data then obtained.

In order to ensure a high efficiency it is important that the entrance to a nozzle be very easy and well-rounded. With a nozzle

of the type shown in Fig. 9, Rateau found that weight discharged per second was indistinguishable from its theoretical amount when the external pressure had its critical value. With smaller drops of pressure the discharge, however, fell off, the defect from the theoretical quantity being greater the less the pressure drop. In all probability the jet, under these latter conditions, was forming a *vena contracta,* and thus the effective area of flow was less than that of the discharge end of the orifice. Both these conclusions are supported by experiments made by Stodola with a convergent-divergent nozzle. Here, when the inlet and discharge conditions were those for which the nozzle was designed, the discharge was practically equal to the theoretical, assuming that the pressure in the throat was 0.577 of the initial pressure. When the pressure at the discharge end was increased, thus lowering the total drop of pressure, a *vena contracta* was formed inside the throat.

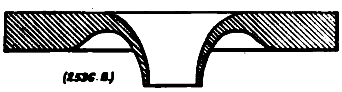


Fig. 9. Rateau's Nozzle.

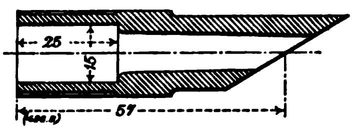


Fig. 10. Briling's Nozzle.

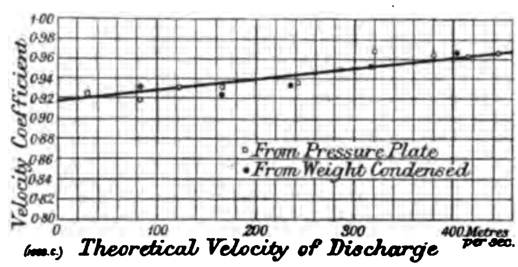


Fig. 11.

Experience with actual turbines has further demonstrated that a *vena contracta* will be formed even when the discharge pressure has its proper value, if the entrance to the throat of the nozzle is not very easy and well rounded, and the discharge may then amount to only some 80 or 90 per cent of the theoretical. The formation of such a contracted vein, moreover, sets up eddies and waves inside the nozzle, thus increasing the losses by fluid friction.

In some experiments on a round nozzle, described by Dr. Briling in the "Zeitschrift Vereines Deutscher Ingenieure," vol. i., 1910, there was found a substantial loss even at the critical velocity of discharge. The nozzle is shown in Fig. 10, and the coefficient of velocity is plotted in Fig. 11. The nozzle, it will be seen, had a very gradual taper, so that the jet in all probability completely filled the orifice at the point of discharge, and did not form a contracted vein

beyond it. On the other hand, however, the entrance to the nozzle is rather abrupt, a contracted vein being presumably formed there, and this, perhaps, accounts for the fact that at the critical velocity the velocity coefficient is about 0.97 instead of unity, as found by Rateau. It should, moreover, be added that some doubt attaches to the values given by the curve for the lower velocities of flow. In such conditions the quantities to be measured are themselves

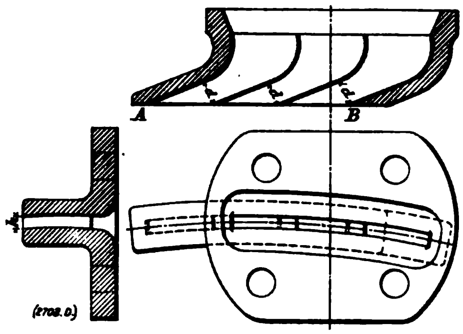


Fig. 12. Christlein's Guide Blades.

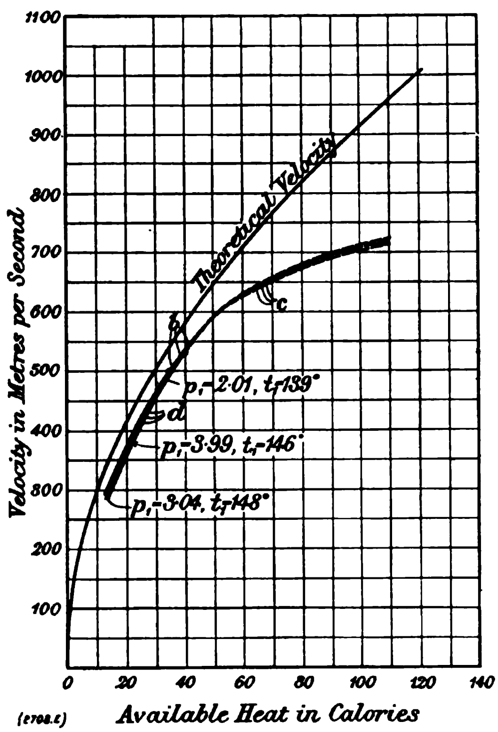


Fig. 13.

small, and any errors of measurement become proportionately great. It would be reasonable *a priori* to expect nearly unit efficiency with very low velocities, since Osborne Reynolds has shown that in the case of pipes the friction losses, at very low speeds, vary only as the first power of the speed instead of as the square. Hence there are reasonable grounds for supposing that the curve for the velocity coefficient should not be a straight line.

Straight nozzles of the types shown in Figs. 9 and 10 are not generally used in turbine practice. The more usual form is represented in Fig. 12, above, whilst a form used for high pressure drops by the Allgemeine Electricitats Gesellschaft (A.E.G.) is similar to that represented in Figs. 20, 21, and 22, page 24.

The curvature appears to have an adverse effect on the efficiency, and the coefficient of velocity is accordingly much lower than might be expected from the results obtained with straight nozzles.

Experience with actual turbines shows that when the speed

of efflux is 800 ft. to 1200 ft. per second, the coefficient of velocity does not exceed some 92 per cent., corresponding to a loss of about 15 per cent. of the energy due from the steam.

Some experiments made by Dr. Paul Christlein ("Zeitschrift Vereines Deutscher Ingenieure," vol. i., 1912) with the guide blades illustrated in Fig. 12, gave even lower values for this coefficient.

The maximum efficiency found by Dr. Christlein, with guide blades of the type in question, was about 90 per cent. for velocities in the neighbourhood of 1900 ft. to 2100 ft. per second, the corresponding velocity coefficient being 95 per cent., and to obtain this figure it was necessary to superheat the steam by about 130 deg. Fahr. With steam having only some 4 deg. Fahr. superheat, the best figure recorded was a velocity coefficient of 0.915, corresponding to an efficiency of only 84 per cent. for an estimated velocity of outflow equal to about 1770 ft. per second.

No measurements were made with low velocities of flow, but, as shown in Fig. 13, an actual velocity of 300 metres, or about 1000 ft. per second, corresponded to, in one case, only about 0.84 of its theoretical value. The corresponding loss in the nozzle was therefore about 30 per cent.

In Fig. 13 the ordinates represent the speeds of efflux in metres per second, and the abscissæ the available heat in calories. The upper curve denotes the theoretical velocity, and the curves below show the actual measured velocities, with initial pressures of 2.01, 3.99, and 3.04 kg. per sq. cm. The initial temperature of the steam in degrees Centigrade is annexed in each case. The superheat corresponding to t_1 was 2 deg. Cent., to t_2 and t_3, 18 deg. Cent. The slope of the curves rather appears to indicate that at low velocities of 300 ft. to 400 ft. per second a better coefficient would be obtained. With steam speeds of 800 ft. to 1200 ft. per second, the results obtained with actual turbines appear, as already stated, to indicate a nozzle efficiency of about 85 per cent. for blades of patterns similar to those in Fig. 12. This efficiency corresponds to a velocity coefficient of about 0.92.

Having determined the actual velocity of discharge by multiplying the theoretical velocity by the above velocity coefficient, the area required to pass, per second, w lb. of steam is given by the relation

$$w V = \Omega v,$$

where V denotes the specific volume of the steam on discharge, Ω the area through the nozzles in square feet, and v the actual velocity of efflux. It is always assumed that guide blades of the type in question run full bore, hence the area between an adjacent pair of blades is equal to the radial width of the blades, multiplied by the shortest distance between them. It is further assumed that the steam leaves the blades at the same angle as that to which the tails are inclined, which, in the case of Fig. 12, is 20 deg. So far as the author is aware, no direct measurement has been made of the actual angle of outflow; but the assumption in question is probably nearly correct for blades of the form in question. If pieces of cotton are strung through blades of this kind, their motion indicates that the steam is delivered in a somewhat turbulent condition.

If α be the angle of discharge, and A B the total arc subtended by the blades, the area through the blades can also be expressed in the form

$$\Omega = F \,.\, A\,B \times h \sin \alpha,$$

where F is known as the thickness coefficient, and h, as in Fig. 12, denotes the radial height or depth of the blades. If the blades were infinitely thin, F would be unity. In practice it ranges from 0.92 to 0.94.

Another form of guide blade, which is that generally adopted for the Parsons steam turbine, is illustrated in Fig. 14. No direct experiments on the efficiency of such blading have been published, but from the equations established, in a subsequent Chapter, for the flow through a group of Parsons blading, it is possible, from the turbine efficiencies observed and the weight of steam passed per second, to calculate the efficiency of the blades if used simply as nozzles. An application of the method of least squares to such data gives a nozzle efficiency of about 0.90 for velocities of 200 ft. to 300 ft. per second. This corresponds to a velocity coefficient of about 0.95. Fig. 14 represents the normal form of Parsons blading. Another form, known as a wing blade, is represented in Fig. 15.

Owing to the fact that these blades have round backs, the area available for flow between adjacent blades is no longer given by the relation $\Omega = d\,h$, where d is the shortest distance between the blades and h the blade height. Thus if, as is usual, d is gauged

to be $\frac{1}{3}$ the pitch and l denotes the total length subtended by the blades, the actual area available for flow is not $\frac{1}{3}\ l h$, but is equal to $l h \sin a$, where a is the angle of the discharge. The thickness coefficient for blades of this type is unity.

A very ingenious method of measuring the angle of discharge for Parsons blading has been adopted by Messrs. W. Chilton and J. M. Newton of the Brush Electrical Engineering Company, Limited, Loughborough. A number of blades were assembled in a block, and

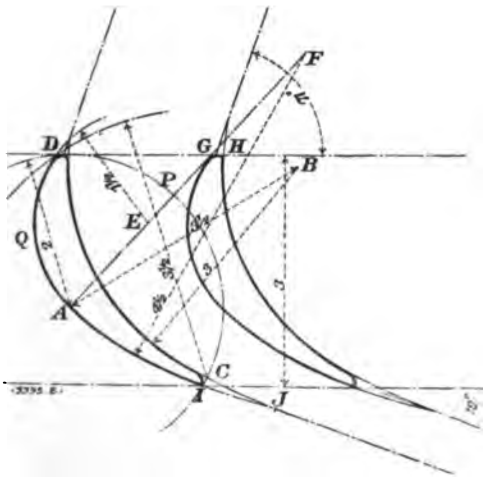


Fig. 14. Normal Parson's Blading.

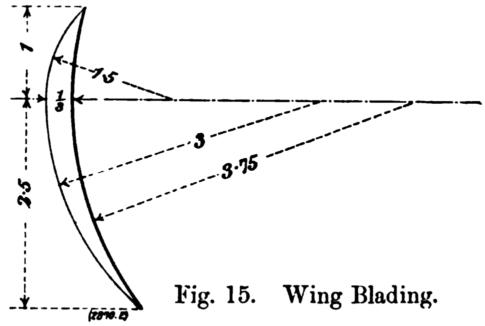


Fig. 15. Wing Blading.

Fig. 16.
Flow of Steam Through Parsons Blading.

carefully set to pitch and gauged. Steam was then passed through the blades into the atmosphere, with a pressure drop of about 1 lb. Pieces of black cotton strung through the blades then gave the angle of flow, which could be measured with certainty to within less than 1 deg. of arc. The arrangement used is illustrated in Fig. 16.

It is usual to set the blades in the casing of a Parsons turbine to a wider pitch than those of the rotor. In the former case the pitch for $\frac{3}{8}$-in. blades is $\frac{1}{4}$ in., and in the latter $\frac{3}{16}$ in. In each case the blades are set so that the narrowest part of the opening between two blades is equal to one-third the pitch. On test the two arrangements were found to have different discharge angles.

Somewhat paradoxically, this angle is smallest in the case of the more widely-pitched casing blades, where for the $\frac{3}{8}$-in. standard it lies between 17 deg. and 18 deg. The $\frac{3}{8}$-in. rotor blades, on the other hand, have a discharge angle between 18 deg. and 19 deg. The best average value for the two sets of $\frac{3}{8}$-in. blades appears to be about $18\frac{1}{3}$ deg. The experiment further showed that the discharge from blades of this type was fairly free from turbulence.

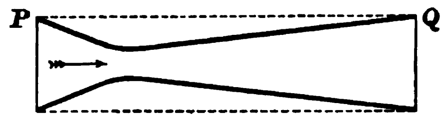


Fig. 17.

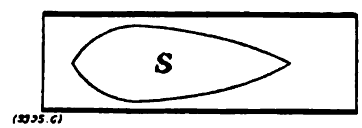
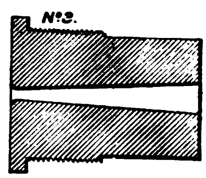


Fig. 18.



SECTION OF NOZZLE

Fig. 19. Typical Nozzle.

Fig. 20.

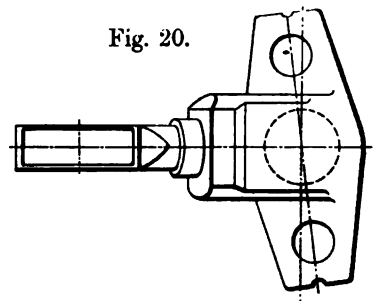


Fig. 21.

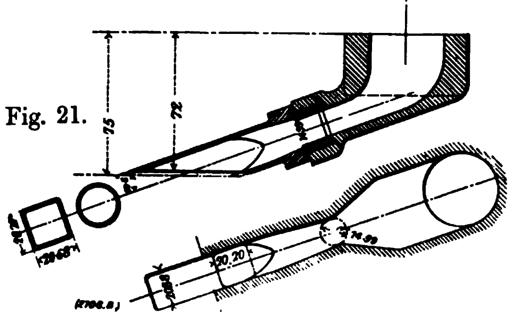


Fig. 22. Christlein's Nozzle.

The reason why no thickness coefficient is needed in the case of guide blades of the type in question is that the stream lines from opposite sides of a blade unite, as indicated by C J I, Fig. 14. The form of the blades is a form of least resistance, and were it not for the curvature the efficiency would undoubtedly be very high. Thus if we take the well-known Venturi meter represented diagrammatically in Fig. 17, then, in spite of the narrow channel afforded to the water at the throat, the loss of head between P and Q, corresponding to a given discharge, is practically identical with what it would be if the pipe were of uniform bore, as indicated by the dotted lines. Obviously the conditions will not be essentially changed if inside a straight pipe we insert a torpedo-shaped body, as indicated in Fig. 18. In this case, as in the other, the discharge under a given head will depend almost wholly on the full bore of the pipe, and

hardly at all on the narrow annular channel round the body S. Of course, this will not hold in extreme cases, as, for example, with a $\frac{1}{8}$-in. throat in a 6-in. pipe; but under reasonable conditions the presence of a body of fair form causes very little loss of head.

The curvature of the blades has some effect on these conclusions, but the type shown in Fig. 14 was originally arrived at by Mr. Horatio Phillips, who found that, as applied to aeroplanes, it showed a much higher efficiency than blades of the type represented in Fig. 12.

Experiments made by Lewicki and Stodola with straight convergent-divergent nozzles, such as represented in Fig. 19, show that the loss of available energy per cent. may, in the case of non-superheated steam, be about $6\left(\frac{u-45}{100}\right)$, when the entrance to the nozzle is easy and well-rounded and the taper not too rapid. Here u denotes the available heat measured from the Mollier diagram. Thus if u is 145 B.Th.U., the nozzle loss will be about 6 per cent., or 8.7 units of heat, so that the actual velocity of efflux will be given by

$$\begin{aligned} v &= 224\sqrt{136.3} \\ &= 2610 \text{ ft. per second nearly.} \end{aligned}$$

In a great many cases, however, turbines have been constructed with nozzles to which the steam does not get ready access; the curves leading to the throat being much too abrupt, the discharge is then less than the theoretical, and the frictional losses are greatly increased.

Nozzles of the pattern represented in Figs. 20, 21, and 22 have been somewhat extensively used. As tested by Dr. Paul Christlein ("Zeitschrift Vereines Deutscher Ingenieure," vol. i., 1912), the maximum efficiency observed was somewhat low, the losses being fully twice as great as by the above formula. Some of his results are plotted in Fig. 23, page 26.

Methods of Testing Nozzles and Guide Blades.

One method of testing the efficiency of a nozzle, originated by Dr. Stodola, is represented in Fig. 24. A fine tube, $\frac{1}{10}$ in. in outside diameter, passes through the nozzle G, as indicated. This tube is closed at its lower end, but is pierced with a hole, $\frac{1}{32}$ in. in diameter, in the part where it passes through the nozzle. The

upper end communicates with a pressure gauge B, which records the pressure of the steam at the $\frac{1}{32}$-in. hole. Provision is made for

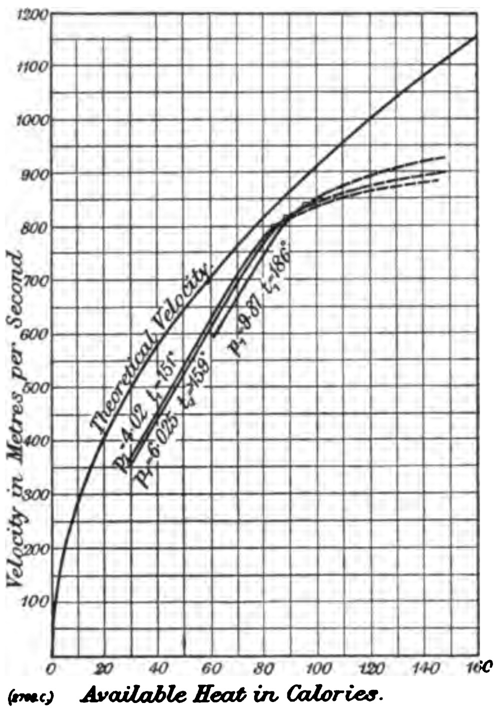
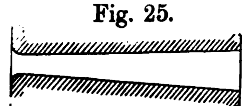


Fig. 23.

Fig. 25.



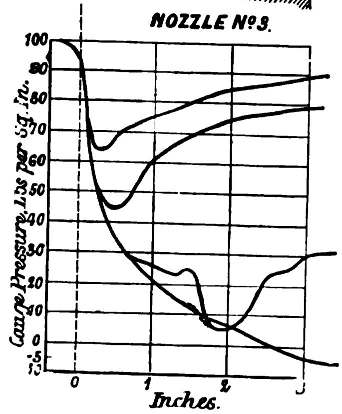


Fig. 26.

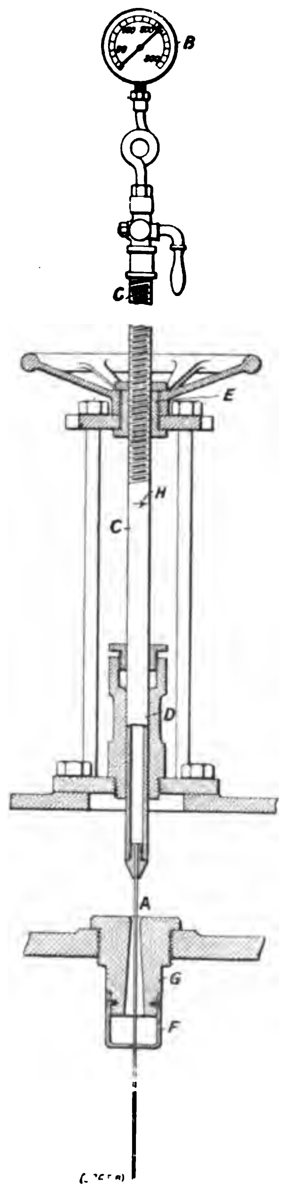


Fig. 24.
Morley's Apparatus for Nozzle Testing.

moving this hole to any part of the nozzle under test, so that the steam pressure at any section of the nozzle can be determined. In this way the whole fall of pressure as the steam passes through

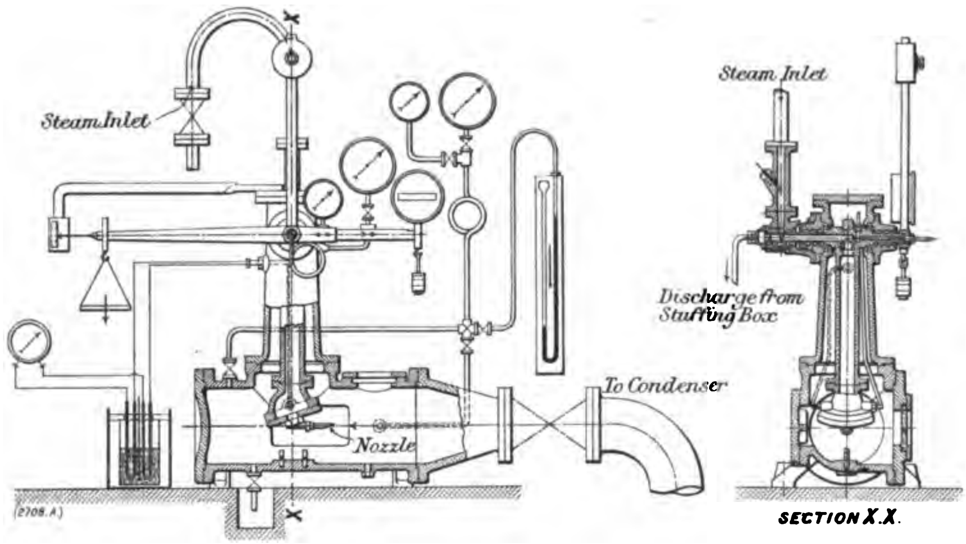
the nozzle can be mapped out. Some results obtained by Mr. Thomas B. Morley, B. Sc., with the nozzle shown in Fig. 25, are represented by the curves in Fig. 26. In the case of the upper curves of Fig. 26, the back pressure on the nozzle was too high. In such a state of affairs a *vena contracta* forms inside the nozzle, which consequently does not run full bore in this neighbourhood. At this *vena contracta* the pressure at first falls too low, and then is gradually restored, as indicated. In the case of the fourth curve, the back pressure is nearly correct for the nozzle, and the pressure, it will be seen, falls pretty uniformly from the throat to the discharge end.

In testing a nozzle by means of this exploring tube, the quantity of steam passed in a given time is weighed, and since the pressure at each section of the nozzle is measured, the actual velocity of flow there can be calculated, assuming the dryness fraction of the steam. From the Mollier diagram the velocities corresponding to any assumed dryness fractions can be calculated by methods such as were explained in the last Chapter, and by comparing these calculated velocities with those corresponding to the assumed dryness fractions, the true dryness fraction can be determined. A reference to the Mollier diagram then shows how much of the heat drop corresponding to the actual fall of pressure has gone to dry the steam, and this represents the loss by friction. In this way Stodola found that with a nozzle supplied with steam at about $10\frac{1}{2}$ atmospheres, and discharging against a back pressure of about $2\frac{3}{4}$ lb. absolute, the loss amounted to about 15 per cent.

Attempts have also been made to measure by electrical means the temperature of the steam at various points along the axis of a nozzle. Were this done the corresponding pressure could be taken from a steam table, and the efficiency determined as above. It has not, however, proved feasible to measure accurately the true temperature of a rapidly-moving gas. Thus Sir C. A. Parsons and Mr. G. Stoney compared the luminosity of a platinum wire inside a chamber (where gas was burnt under pressure) with the brightness of a similar wire placed in front of a nozzle through which the products of combustion expanded. The two wires appeared to have practically the same temperature. The gas, cooled by expansion through the nozzle, had its temperature restored by adiabatic compression as it impinged on

the wire in its path. Other experiments on steam turbines have also shown the extreme difficulty—it might perhaps even be said the impracticability—of accurately measuring the temperature of the steam at various points along the turbine.

A more direct method of measuring the efficiency of a set of guide blades or of a nozzle appears to have been first used by Mr. D. Napier, and was adopted in the experiments of Dr. Christlein ("Zeitschrift Vereines Deutscher Ingenieure," vol. i., 1912) already referred to.



Figs. 27 and 28. Christlein's Nozzle-Testing Apparatus.

In this method the back thrust or reaction of the escaping jet is measured, and the velocity of efflux deduced therefrom. The apparatus, as used by Dr. Christlein, is represented in Figs. 27 and 28. It consists of a pendulum pipe supported on ball-bearing trunnions, as indicated. The nozzle under test is fixed at the lower end as shown, and by its back thrust tends to deflect the pendulum. This deflection is counteracted by weights in the scale pan. The steam which escapes from the nozzle is collected in a surface condenser and weighed.

If w be the weight passed per second, and R the measured reaction in pounds, the velocity of efflux v is given by the relation

$$R = \frac{w}{g} \cdot v.$$

This equation, in one form or another, is a fundamental one in

steam-turbine theory, and, indeed, in mechanics generally. The quantity on the right is the momentum generated in w lb. of steam in one second, and this is numerically equal to the force effecting the change in the velocity.

As applied to determining velocities of efflux, the equation in question gives rise to a somewhat curious point when used for the case of a simple convergent nozzle with which, as the external pressure is lowered, the velocity of efflux increases up to the critical value. Once this value is attained any further decrease in the external pressure will not increase the velocity of *efflux*, but it will increase the measured reaction, and there is indeed some corresponding increase of onward velocity, which is, however, generated after the steam has actually cleared the nozzle. At the critical point the distribution and amount of the pressure on the interior walls of the nozzle box attain conditions which then remain contant, however much the external pressure be lowered.

Let A be the area of the back wall of the nozzle box. Then the pressure on the inner side of this is p_0 A. On the front face the presence of the nozzle opening prevents the internal pressure being uniformly distributed, and the consequence is that the net effective pressure on this face is equal to p_0 A^1, where A^1 varies with the external pressure until the critical value is reached. After that A^1 remains constant.

If Ω be the area of the nozzle opening, the outer pressure p_1 acts on the whole of the back face, but on the front the effective area exposed to pressure is less by the amount of the nozzle opening. Hence the total reaction observed consists of two terms, or

$$R = p_0 (A - A^1) - p_1 \Omega.$$

When the critical condition is reached, A^1 becomes constant, but the term $p_1 \Omega$ diminishes with any further reduction in p_1, so that the reaction is increased, although the velocity of *efflux* remains unchanged.

As discharged the steam has, however, a pressure greater than that of its surroundings, and a further expansion takes place both laterally and in a forward direction, and it is this addition to the forward velocity produced after the jet is actually clear which corresponds to the observed increase of reaction.

CHAPTER IV.

IMPULSE BUCKETS.

THE function of the guide blades being to convert potential into kinetic energy, that of the buckets on the moving wheel is to transfer this energy to the turbine shaft by robbing the steam of its velocity. In practice, this function is only imperfectly accomplished. The fluid as finally discharged has always some residual velocity, and hence some remanent kinetic energy. There are, besides, certain losses by fluid friction, so that in practice the wheel does not transfer to the shaft more, at most, than some 80 per cent. of the kinetic energy delivered to it by the nozzles. If the latter have an efficiency of 85 per cent., the over-all efficiency of the combination of nozzles and moving wheel is about $0.85 \times 0.80 = 0.68$, which is about the maximum figure attained with non-superheated steam. Superheating, it may be stated, diminishes the frictional losses both in guide blades and buckets.

Impulse Buckets.

The elementary theory of the moving impulse bucket is very simple. In Fig. 29 is represented diagrammatically the nozzle of a De Laval turbine, with the buckets below it. Steam is delivered from the nozzle in a definite direction, and with a definite velocity, which can be calculated by the rules already given.

In Fig. 30 let $AB = v$ denote this velocity drawn to scale. The angle a is the angle which the direction of flow from the nozzle in Fig. 29 makes with the direction of motion of the blades. CB (Fig. 30) denotes, both in magnitude and direction, the speed of the buckets; and then AC is equal to the velocity of the fluid measured "relatively" to the moving buckets. In other words, if a flat surface were moving with the velocity CB, then a particle moving simultaneously over this surface with a velocity AC would

actually travel in space along the path A B, so that A B is the "absolute" velocity in space of the particle in question.

One of the buckets of Fig. 29 is represented by the heavy line H I J in Fig. 30. This catches the steam moving along the relative path A C, and turns it round, so that it is discharged along the relative path C D; C D being a tangent to the curve of the bucket at discharge. This, as stated, is a relative velocity, and to find the true absolute direction of the discharged steam in space it is necessary to add (graphically) to this relative velocity C D (as determined later), the bucket speed D E = C B = s. Then C E = r denotes the absolute residual velocity of the steam in space.

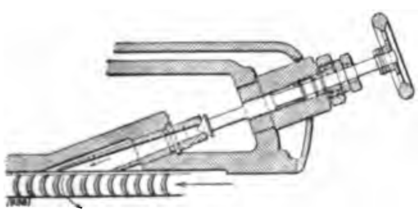
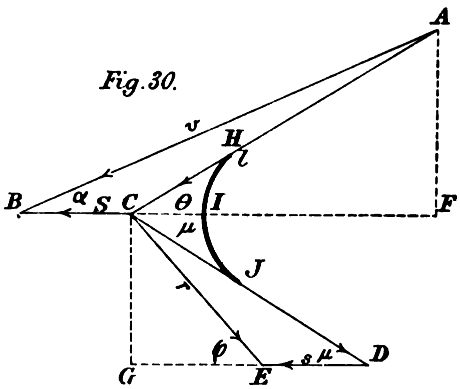


Fig. 29. De Laval Nozzle and Buckets.

Fig. 30.



Hence 1 lb. of the steam entering with an absolute kinetic energy $= \left(\frac{v}{224}\right)^2$ B.Th.U. is discharged with an absolute kinetic energy of $\left(\frac{r}{224}\right)^2$ B.Th.U. per lb. These kinetic energies can, of course, also be expressed in foot-pounds, viz., as $\frac{v^2}{2g}$ and $\frac{r^2}{2g}$.

Hence the steam has lost in passing through the wheel an amount of energy equal to $\left(\frac{v}{224}\right)^2 - \left(\frac{r}{224}\right)^2$. Of this, however, only a part has been transmitted to the shaft as useful work, since there is always some loss in friction due to the rubbing of the steam over the buckets. As a consequence of this, the relative velocity C D with which the steam leaves the bucket is equal to ψ A C = $\psi\, l$, where ψ is a fraction which varies considerably under different conditions. For simple impulse wheels, running at about their best

speeds, ψ is often about 0.68 to 0.72. The loss of kinetic energy in the wheel is $\left(\frac{l}{224}\right)^2 (1 - \psi^2)$ B.Th.U.

Hence of the original kinetic energy supplied to the wheel—viz. $\left(\frac{v}{224}\right)^2$ B.Th.U., an amount equal to $\left(\frac{r}{224}\right)^2$ has been "carried over," and an amount $\left(\frac{l}{224}\right)^2 (1 - \psi^2)$ has been destroyed in friction. The residue—viz., $\left(\frac{1}{224}\right)^2 [v^2 - r^2 - l^2 (1 - \psi^2)]$, is the "indicated" work done on the turbine shaft. The actual useful work delivered to the generator, or whatever is driven by the turbine, is always less than this, owing to bearing friction and the like. The wheel efficiency, or, as it has been conveniently termed by Mr. Jude, the diagram efficiency, is equal to the indicated work done per pound of steam, divided by the original kinetic energy of the jet. Calling this e, we get

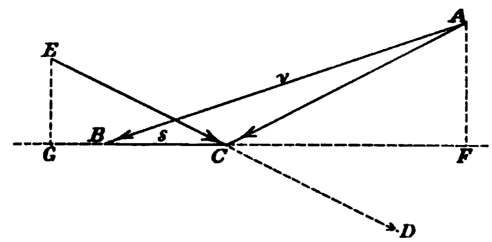


Fig. 31.

$$e = \frac{v^2 - r^2 - l^2 (1 - \psi^2)}{v^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

If we know ψ, both l and r are easily scaled off the diagram (Fig. 30), and thus e may be determined. This, however, can be done even more simply, as indicated in Fig. 31.

As already stated, C D, Fig. 30, is directed along the tangent, at discharge, to the curve of the bucket. The angle F C D = μ (Fig. 30) is known as the bucket angle at discharge. This in simple impulse wheels is generally 25 deg. to 30 deg. In Fig. 31 the angle F C D is made equal to μ, but instead of setting off C D = ψl below the base line, this quantity is set off above the line as E C. From E a perpendicular is dropped to the base line at G, and similarly A F is drawn perpendicular to the same base line. If, then, the quantity F G is scaled off the diagram, the diagram efficiency e is given by the relation

$$e = \frac{2\,\mathrm{CB}\,.\,\mathrm{FG}}{\mathrm{AB}^2} = \frac{2s\,.\,\mathrm{FG}}{v^2} \quad . \quad . \quad . \quad . \quad . \quad (11)$$

Or if $s = \delta \,.\, v$; δ being a fraction, we have

$$e = 2\delta\,.\,\frac{\mathrm{FG}}{v} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

As a numerical example: Let A B be 1400 ft. per second, s = 450 ft. per second. Let the nozzle angle be 20 deg., and the bucket angle on discharge 28 deg. Then setting off to scale A B and C B, we get A C = l = 989.2. Taking ψ as 0.72 we get E C = 712.2, and the angle E C G being the angle of discharge is, as stated, 28 deg. Scaling G F, it will be found to be 1494 ft. per second. Hence

$$e = \frac{2 \times 450 \times 1494}{1400 \times 1400} = 0.686$$

The diagram efficiency may also be found by calculation instead of graphically.

From Fig. 31, we have, if α denote the angle A B F, and μ the angle E C G

$$\text{G F} = v \cos \alpha + \psi\, l \, . \cos \mu - s$$

so that

$$e = \frac{2\, s}{v^2} \Big[v \cos \alpha + \psi\, l \cos \mu - s \Big]$$

$$= \frac{2 \times 450}{1400^2} \Big[1400 \times 0.9397 + 0.72 \times 989.2 \times 0.8829 - 450 \Big] = 0.686 \text{ as before.}$$

On actual trial it will be found that the equations (10) and (11) yield the same value for e.

The direct proof of (11) is as follows:—If w be weight of a body moving with a velocity v, its momentum is equal to $\frac{w}{g} v$. Now v can always be resolved into two components. Thus in Fig. 30 the velocity A B can be resolved into the two components A F and F B. The former is called the axial velocity of the steam, whilst F B is known as the tangential velocity. Taking w as unity, we have $\frac{v}{g}$ for the total momentum of 1 lb. of steam as it issues from the nozzles, and this can be resolved into an axial momentum $\frac{\text{A F}}{g}$ and a tangential momentum $\frac{\text{F B}}{g}$, which is directed in the same direction as the motion of the buckets. Similarly, if we take the final absolute velocity r, the total momentum of 1 lb. of steam on discharge is $\frac{r}{g}$. This can also be resolved into an axial component $\frac{\text{C G}}{g}$ and a tangential component $\frac{\text{G E}}{g}$. Hence the tangential momentum of 1 lb. of steam in passing the buckets has been

changed by the amount $\frac{BF + GE}{g}$. This quantity is frequently called the "Impulse" of the steam on the bucket.

By an elementary rule in mechanics, if the momentum in a given direction of a body weighing w lb. be altered, then the force f, which has effected this alteration, is given by f = change of momentum effected in one second. Hence the force which has changed the tangential momentum of 1 lb. of steam in one second is

$$f = \frac{BF + GE}{g}.$$

This, then, is the tangential force exerted by the buckets on the steam, and, action and reaction being equal and opposite, a force of equal amount must have acted on the buckets. In one second this force drives the buckets through a distance s. Hence the indicated work done per second per pound passed is

$$fs = \frac{s.(FB + GE)}{g}.$$

The kinetic energy supplied per pound per second is $\frac{v^2}{2g}$.

Hence

$$e = \frac{s}{g}(FB + GE) \div \frac{v^2}{2g} = \frac{2s(FB + GE)}{v^2}.$$

From Fig. 30 we have $FB = v \cos \alpha$; $GE = \psi . l . \cos \mu - s$. So that, as before,

$$e = \frac{2s}{v^2}.(v \cos \alpha + \psi . l . \cos \mu - s) \quad . \quad . \quad . \quad (13)$$

It may be noted in passing that unless A F is equal to C G, Fig. 30, there is also a change of axial momentum, and thus an axial force on the bucket. Hence an axial thrust may arise, even if there is no drop in the pressure of the steam as it passes from one side of the wheel to the other.

From equation (13) it will be seen that the diagram efficiency depends on s. If s be zero, the bucket being stationary, no work is done, and the value of e is also zero.

Again, if $s = v \cos \alpha + \psi . l \cos \mu$, the efficiency is again zero, so that at some intermediate point, the diagram efficiency must be a maximum. It is found by experiment that the position of this maximum depends not only on the nozzle angle α, but also on the bucket angle at discharge. The best value of s is approximately one-half F B.

Table II. shows the efficiencies realised in a series of tests of six turbine wheels, the nozzle angle in all experiments being 20 deg., whilst the bucket angles varied between 20 deg. and 50 deg. The figures tabulated have been deduced by smoothing out the actual trial data by the method of lead squares.

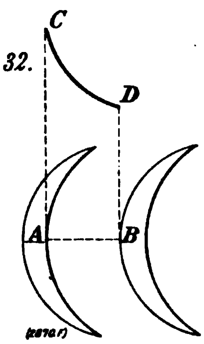
TABLE II.—EFFICIENCY OF IMPULSE TURBINES.

$\frac{\text{Blade Speed.}}{\text{Steam Speed.}}$	Wheel Efficiencies at Different Ratios of Blade Speed to Steam Speed, for Different Bucket Angles. Nozzle Angle = 20 Deg.									
	Bucket Angle = 20 Deg.									
	0	1	2	3	4	5	6	7	8	9
.0	.0000	.0334	.0642	.0954	.1261	.1560	.1853	.2145	.2425	.2701
.1	.2970	.3231	.3489	.3737	.3988	.4214	.4432	.4663	.4878	.5513
.2	.5286	.5476	.5665	.5843	.6014	.6179	.6334	.6485	.6625	.6758
.3	.6884	.7001	.7109	.7208	.7297	.7380	.7457	.7525	.7592	.7647
.4	.7699	.7751	.7799	.7852	.7903	.7959	.7991	.8009	.8000	.7960
	Bucket Angle = 30 Deg.									
.0	.0000	.0318	.0631	.0940	.1222	.1542	.1841	.2133	.2418	.2695
.1	.2964	.3232	.3484	.3732	.3969	.4197	.4426	.4642	.4852	.5056
.2	.5253	.5443	.5623	.5798	.5964	.6123	.6276	.6416	.6550	.6678
.3	.6797	.6908	.7006	.7094	.7178	.7250	.7318	.7379	.7434	.7483
.4	.7526	.7562	.7594	.7619	.7640	.7654	.7665	.7669	.7670	.7665
	Bucket Angle = 40 Deg.									
.0	.0000	.0320	.0633	.0939	.1212	.1531	.1815	.2094	.2369	.2633
.1	.2894	.3148	.3380	.3629	.3860	.4084	.4294	.4510	.4707	.4908
.2	.5096	.5277	.5449	.5614	.5773	.5923	.6066	.6200	.6329	.6451
.3	.6560	.6663	.6758	.6844	.6922	.6990	.7053	.7104	.7160	.7204
.4	.7241	.7271	.7295	.7314	.7326	.7332	.7332	.7328	.7319	.7303
	Bucket Angle = 50 Deg.									
.0	.0000	.0308	.0609	.0903	.1189	.1469	.1742	.2008	.2225	.2515
.1	.2759	.2995	.3224	.3449	.3664	.3873	.4075	.4277	.4457	.4639
.2	.4813	.4980	.5141	.5293	.5440	.5579	.5720	.5837	.5955	.6067
.3	.6172	.6270	.6363	.6449	.6527	.6600	.6665	.6721	.6770	.6813
.4	.6847	.6873	.6892	.6904	.6907	.6903	.6890	.6869	.6840	.6803

It has been shown above that it is possible in a very simple way to determine the driving force on a bucket, once the value of ψ is known. It is both of interest and importance to note that this driving force really arises from centrifugal action, as in this appears to lie the explanation of the enormous variations in the value of ψ which have been disclosed by experiment.

Whenever a body is compelled to move round a curved path, centrifugal forces necessarily come into play. The steam as it passes round the concave face of the bucket in Fig. 30 is no exception to this rule. Were the bucket absent, it would continue its original motion in a straight line, and to compel it to follow a curved line, the concave face of the bucket must exert on it a certain pressure. Hence the layer of fluid next this concave face is under pressure, the state of affairs being somewhat as indicated in Fig. 32, where the ordinates to the curved line C D represent the distribution of pressure along the base A B.

Fig. 32.



The pressure of the fluid at the concave face is equal to A C, whilst at B it is represented by B D; B D being probably the same as the pressure of the fluid as it was originally delivered from the nozzle. The diagram, it should be stated, is intended to be diagrammatic only, and is not to scale, since the data do not exist to find the true distribution of pressure in the case of a viscous fluid like steam. Were the latter a perfect fluid the pressure distribution could be calculated, and the condition of affairs in that case affords some clue as to the actual character of the phenomena in the case of steam.

If we suppose that each layer of fluid remains distinct during its passage through the bucket, what is known as stream line motion results. In that case each individual layer retains the same energy as that with which it started from the nozzle. Hence, as the layer next the concave face is higher in pressure than layers further out, its velocity of flow will be decreased. At the same time being under pressure as stated, and unconfined laterally, it will spread out sideways, with a velocity corresponding to its surplus of pressure. There thus results a spreading of the jet. The rate at which this takes place may, in certain cases, equal the speed of flow in the forward direction.

The thinning of the jet when unconfined laterally is very marked when a nozzle delivering steam at a high velocity is directed on to a Pelton bucket. The jet, originally $\frac{3}{8}$ in. or so in diameter, is then delivered over the discharge edges of the bucket in sheets as thin as paper. The boundaries of the stream have thus a considerable lateral velocity, which contributes nothing to the drive on the wheel.

When the stream is unconfined laterally it is said to pass through the buckets with free deviation. Values of ψ for buckets working with free deviation are given in the curve, Fig. 33.

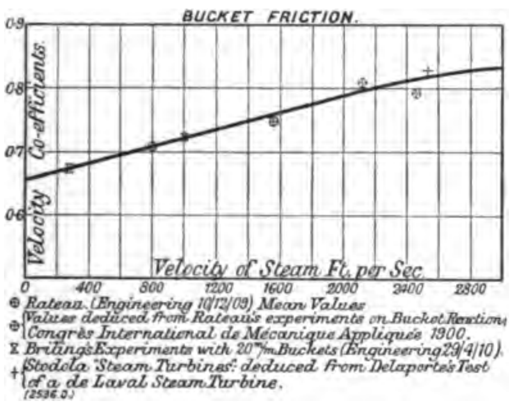


Fig. 33.

Experience has shown, however, that higher efficiencies (sometimes much higher efficiencies) are obtained by preventing this lateral spreading of the jet. In fact, it is found advantageous to shroud the blading even when the steam velocity is relatively low, in which case the spreading of the jet is not excessive. Thus the earlier Zoelly turbines had buckets much longer radially than their guide blades, so that free deviation was secured. This was, however, subsequently abandoned in favour of Rateau's plan of making the buckets but little wider than the guide blades. Many earlier turbine builders were afraid that unless the bucket was much wider than the nozzle there would ensue losses by "spilling," the jet passing round in place of through the buckets. In some experiments made by Messrs. Chilton and Newton, of the Brush Electrical Engineering Company, Loughborough, no signs of spilling could, however, be detected with a bucket only a few mils. wider than the jet. The latter was, however, round, and it is quite probable that a somewhat larger allowance would be necessary with guide blades giving a sheet rather than a round jet of steam. With round jets a large amount of free deviation is unavoidable, and their use is accordingly contra indicated.

In other words, restraining the lateral spreading of the jet increases the value of ψ, but the latter is also dependent on the "angle of deviation" of the jet. This angle is equal to $180 - \theta - \mu$,

Fig. 30. In this figure it should be noted that θ is the angle A C F, which is not necessarily the bucket angle at entrance. In fact, the latter angle should always be some 5 deg. to 10 deg. larger than θ.

From some experiments by Dr. Briling ("Zeitschrift Vereines Deutscher Ingenieure," April 29, 1910) the following expression was deduced for the connection between ψ and ω, the angle of deviation as above defined:—

$$\psi = \psi_0 - 0.000,432\ \omega^{\frac{4}{3}}.$$

The value of ψ_0 was found to vary between 0.90 and 0.99, and the jet was more or less "freely deviated."

Briling also found that buckets (wide in the axial direction) gave larger values of ψ than narrower buckets. This, perhaps, arose from the fact that with the wider buckets there was some restraint on the spreading of the jet. The ratio of width to pitch was, in fact, the same for the narrow and for the wide buckets, so that a greater mass of steam passed between each pair of the latter. To allow the stream to spread equally freely in both cases, the wide buckets should therefore have been longer than the others, but were actually of about the same length. Hence there may well have been with them some restriction on the spreading of the jet. Further, if from Table II., as explained below, the coefficients ψ are calculated for the wheel running at different speeds, it will be found that the slower the speed the higher is the value of ψ. Now the tendency of the jet to spread is greater the greater the drive on the wheel, and this drive is a maximum when the wheel is stationary. Hence it may well be that at the higher bucket speeds the jet was freely deviated, but that at the lower there was some check on its lateral spreading. The matter is, however, complicated by the further fact that the slower the speed the greater the relative velocity of the jet. In some experiments carried out with water by Donat Banki ("Zeitschrift Vereines Deutscher Ingenieure," vol. i., 1909), the lateral spreading of the jet was very effectually prevented, and high values of ψ realised

Hydraulic Shock.

It was at one time considered of prime importance that the tangent to the bucket angle at entrance should be parallel to the

line C A, Fig. 30. In other words, the bucket angle at entrance was made equal to A C F. With this construction it was held that the fluid slid into the bucket rather than struck it, and it was anticipated that this condition would lead to a maximum value of ψ. Actual experience has shown the contrary, and that it is best to make the bucket angle at entrance some 5 deg. to 10 deg. larger than the angle θ, Fig. 30.

From Table II., on page 35, it is possible to calculate the value of ψ corresponding to different values of θ. The buckets had equal entrance and discharge angles. Hence with the 30-deg. bucket there was, at low speeds of motion, a large divergence between θ and the bucket angle. The two coincided when the ratio $\dfrac{\text{blade speed}}{\text{steam speed}} = 0.3472$. At this speed the efficiency was 0.7237, and since

$$e = \frac{2s}{v^2} \cdot (v \cos \alpha + \psi l \cos \mu - s),$$

we easily find that $\psi = 0.759$. On the other hand, with a blade speed equal to $\frac{1}{10}$th the steam speed, the value of ψ comes out as 0.818. In this case the value of θ is about 8 deg. less than the bucket angle at entrance.

The reasoning, on which the erroneous conclusion as to the importance of making the bucket angle at entrance equal to θ was derived, is exceedingly plausible, and seems to have originated in Germany. Let a current of water impinge perpendicularly on to a flat plate. It will obviously spread equally in all directions. Suppose, however, the plate is inclined to the current, as indicated in Fig. 34, where A B denotes the plate and C D = v, the velocity and direction of the impinging current. To determine what would happen in this case the velocity C D was resolved into a component C E parallel to the plate and a component E D perpendicular to the plate. The latter component being normal to the plate, was assumed to produce a uniform spread of the fluid in all directions, so that the centre of gravity of the fluid moved along the plate with a resultant velocity equal to C E. The normal component D E was thus supposed to be "destroyed by hydraulic shock." It was assumed

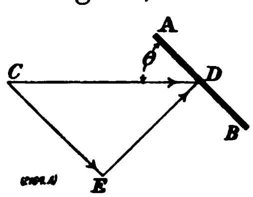


Fig. 34.

that in the case of a turbine bucket the initial element of the latter could be taken as corresponding to the plate A B, and that if this made an angle with the direction of the entering fluid, that there would be a loss corresponding to the component D E in Fig. 34. Actual experiment has shown this conclusion to be erroneous. Experimenting with water, Professor Donat Banki ("Zeitschrift Vereines Deutscher Ingenieure," 1909) found no evidence of serious loss by hydraulic shock, even when the jet was steeply inclined to the surface on which it impinged, but it should be added that any lateral spreading of the flow was, in his experiments, checked by side plates.

CHAPTER V.

THE EFFICIENCY RATIO AND THE REHEAT FACTOR.

IN Chapter II. a method was explained of determining from the Mollier diagram the amount u of heat energy which a theoretically perfect turbine could turn into work when receiving steam at a pressure p_1 and exhausting at a pressure p_2. Thus, taking steam at an initial pressure of 170 lb. absolute and at a temperature of 520 deg. Fahr., the amount of heat which a perfect turbine would turn into work if exhausting at a pressure of $1\frac{1}{2}$ lb. per sq. in. will, as shown by the line H G, Fig. 7, *ante*, be 331 B.Th.U. per lb. of steam passed. The position of the point G further shows that in such a case the steam on discharge would be 83.3 per cent. dry. In all actual turbines there are serious losses due to internal friction, originating in various ways, so that the amount of heat actually converted into work is only some 60 to 70 per cent. of that theoretically possible, even in the case of a good turbine. If the actual figure is 60 per cent., the turbine is said to have an efficiency ratio of 60 per cent. The efficiency ratio may also be defined as

$$\frac{\text{Theoretical steam consumption}}{\text{Actual steam consumption}}.$$

To determine the theoretical steam consumption we note that one horse-power hour is equal to $33{,}000 \times 60 = 1{,}980{,}000$ ft.-lb. Dividing this by 778, we get the equivalent energy in heat units. Hence one horse-power hour = 2545 B.Th.U.

Now if 331 B.Th.U. should be turned into useful work for each pound of steam passed through a perfect turbine, then the theoretical steam consumption per horse-power hour is $\frac{2545}{331} = 7.689$ lb. If on trial the actual consumption is 13 lb. per horse-power hour, then the efficiency ratio of the turbine is 59.1 per cent. Similarly the

theoretical consumption per kilowatt hour may be shown to be $\frac{3412}{u}$ lb. where u denotes the available heat as measured from the Mollier diagram, in the manner already explained. As before, the efficiency ratio will be equal to $\frac{\text{theoretical consumption}}{\text{actual consumption}}$.

The value thus found for the efficiency ratio will depend upon whether the power developed is measured at the switchboard or at the turbine shaft.

Let ϵ_1 denote the efficiency ratio as deduced from the power measured at the switchboard, then ϵ_1 may be called the "over-all" efficiency ratio. If the generator has itself an efficiency of 95 per cent., then ϵ_2, the efficiency ratio at the turbine shaft, is $\epsilon_1 \div 0.95$; and, finally, if the turbine has a mechanical efficiency of 97 per cent., the "indicated" efficiency ratio of the turbine ϵ_3, say, is equal to $\epsilon_2 \div 0.97$. Hence, if ϵ_1 were 68 per cent., then ϵ_2 would be 71.5 per cent., and ϵ_3 would be 73.7 per cent.

It was shown above, that starting with steam at an initial pressure of 170 lb. per sq. in. absolute, and at a temperature of 520 deg. Fahr., and expanding this down to $1\frac{1}{2}$ lb. absolute, the value of u was 331 B.Th.U. If this drop of pressure occurred in a single nozzle, the theoretical speed of efflux would be $v = 224\sqrt{331} = 4074$ ft. per second, and hence to abstract this velocity efficiently it would be necessary to use a wheel with a very high bucket speed.

For this reason it is usual to divide up the whole drop of pressure in a turbine over several compartments or "stages," and this leads, it will be found, to an increase in the "efficiency ratio" attained.

Thus suppose as before that the steam is supplied at 170 lb. absolute, and at 520 deg. Fahr., exhausting at 1.5 lb. absolute. Then, as already stated, the available heat is 331 B.Th.U. per pound. If, as with a De Laval turbine, there is one stage only, with a stage efficiency ratio of 60 per cent., only 198.6 B.Th.U. will appear as useful work on the shaft, the difference going to reheat the steam.

In fact, as the diagram, Fig. 7, shows, the steam had originally a heat content of 1277 B.Th.U. per lb. In the case of a perfect turbine, out of this total energy 331 B.Th.U. would be removed in the form of useful work, the steam being discharged at $1\frac{1}{2}$ lb. absolute, with a heat content of 946 B.Th.U., corresponding, as the

diagram shows, to a dryness fraction of 83.3 per cent. That is to say, each pound as discharged would have entrained in it 0.167 lb. of condensed water. Since the actual turbine only removes $0.60 \times 331 = 198.6$ B.Th.U. per lb., the heat content as discharged at $1\frac{1}{2}$ lb. pressure is $1277 - 198.6 = 1078.4$ B.Th.U., and this, as indicated by J on the diagram, corresponds to a dryness fraction of 96.1 per cent. As dry steam at $1\frac{1}{2}$ lb. pressure has a volume of 227.2 cub. ft. per lb., the actual volume on discharge from a perfect turbine would be $227.2 \times 0.833 = 189.2$ cub. ft. per lb.; whilst in the case of a turbine with a 60 per cent. efficiency ratio it would have a volume on discharge at the same pressure of $227.2 \times 0.961 = 218.2$ cub. ft. per lb.

Now, with the same initial conditions and final pressure, let the total pressure drop be divided up over four compartments. In the first of these the pressure falls from 170 lb. to 68 lb. per sq. in., whilst in the second the exhaust pressure is 24 lb., and in the third 8 lb. absolute. Let the efficiency ratio of each compartment taken separately be the same as before. Then, measuring from the diagram, the theoretical available heat in the first compartment is given by the length of the line H R, which, on measurement, is found to be about 82.5 B.Th.U. The "stage" efficiency ratio being 60 per cent., the actual heat abstracted from the steam in this compartment is, however, only 60 per cent. of this, or 49.5 B.Th.U. per lb. Hence the steam, after doing its work in the first compartment, still contains $1277 - 49.5 = 1227.5$ B.Th.U. Its pressure is 68 lb. per sq. in., and hence the actual condition of the steam is represented by the point S in place of the point R. In the second compartment the steam expands from the point S down to a pressure of 24 lb. per sq. in., and the heat theoretically available is represented by the line S T, which, on scaling, is found to be about 83 B.Th.U. Taking the stage efficiency ratio at 60 per cent. as before, $83 \times 0.60 = 49.8$ B.Th.U. will be removed from the steam in this second compartment, making its heat content, after finishing its work there, equal to $1227.5 - 49.8 = 1177.7$ B.ThU., so that its actual condition on discharge is represented by the point V in place of the point T. In the third compartment the heat theoretically available is equal to V W or 79 B.Th.U., but the actual condition of the steam as it leaves to enter the final compartment is repre-

sented by the point X. In the fourth compartment the theoretical available heat is equal to X Y, or about 105.5 B.Th.U.

Adding together the heat theoretically available in the four compartments, we get a total of 350.0 B.Th.U. as the amount which has been available during the passage of the steam through the turbine, and this, it will be seen, is materially greater than the 331 B.Th.U. which is all that would have been available in a frictionless turbine working between the same initial and final conditions. In short, each compartment delivers the steam to its successor a little drier than it would have done had there been no friction. Of the 350 B.Th.U. which became actually available, 60 per cent., or 210 B.Th.U., were turned into work. Hence the "total" efficiency ratio of the turbine is $\frac{210}{331}$ = 63.4 per cent., although each constituent stage had an individual efficiency ratio of 60 per cent. only. The individual efficiency ratio of each stage may conveniently be called the hydraulic efficiency of the turbine and denoted by η. It will be seen that in the case of a turbine consisting of more than one stage this hydraulic efficiency is always less than the total "indicated" efficiency ratio.

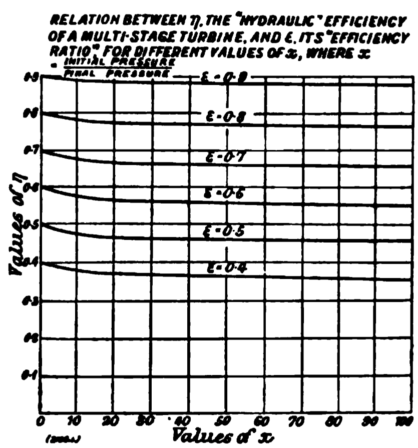


Fig. 35.

In the Chapter on Thermodynamic Principles the relation between the hydraulic efficiency and the efficiency ratio of a turbine is worked out for the case in which there are an infinite number of compartments.

The results are shown in the curves plotted in Fig. 35, where x denotes the ratio of the initial to the final pressure. It will be noted that after an expansion of one hundred-fold or so, the ratio $\frac{\epsilon}{\eta}$ becomes practically constant. This ratio $\frac{\epsilon}{\eta}$ is known as the "reheat factor," and for any turbine working with a considerable range of expansion, the reheat factor η corresponding to any given value of ϵ can, with all necessary precision, be taken from Table III.

TABLE III.

Efficiency Ratios.	High-Pressure Condensing Turbines. Reheat Factor for Different "Indicated" Efficiency Ratios.									
	.00	.01	.02	.03	.04	.05	.06	.07	.08	.09
0.5	1.1006	1.0985	1.0964	1.0942	1.0921	1.0900	1.0880	1.0861	1.0841	1.0822
0.6	1.0802	1.0783	1.0764	1.0746	1.0727	1.0708	1.0689	1.0670	1.0651	1.0632
0.7	1.0613	1.0593	1.0573	1.0554	1.0534	1.0514	1.0494	1.0474	1.0454	1.0434
0.8	1.0414	1.0394	1.0374	1.0353	1.0333	1.0315	1.0297	1.0277	1.0260	1.0250

It was shown above that when the heat available in a perfect turbine was 331 B.Th.U., the heat which actually becomes available in a four-compartment turbine with an hydraulic efficiency of 60 per cent. is 350 B.Th.U. The reheat factor in this case is $\frac{350}{331} = 1.0575$, so that the total efficiency ratio is 0.634. With a single compartment the hydraulic efficiency is the same as the indicated efficiency ratio. The greater the number of compartments the greater is the reheat factor, but there is generally only about 2 per cent. difference between the reheat factor for four compartments and for an infinite number.

A multi-stage turbine should be proportioned not for u units of available heat, but for $\mathrm{R}u$ units where R is the reheat factor. When the number of stages or compartments is small, the pressure and volume of the steam at each intermediate point may be found by the step-by-step process which has been described above, but with many stages this becomes tedious, and, moreover, when each pressure drop is small, it is difficult to measure accurately from the diagram the pressure at each stage corresponding to a given expenditure of available heat. The pressure and volume of the steam at each stage is then most conveniently found by means of the relation $p\,\mathrm{V}^{\gamma} =$ constant, as is explained in detail in the Chapter dealing with the design of an impulse turbine.

CHAPTER VI.

CORRECTION CURVES.

IN turbine tests the actual steam pressure and vacuum are practically never exactly those laid down in the specification, and for which the turbine is designed. Further, a builder may have the drawings and patterns for a particular turbine proportioned to work under stated conditions, and it may then be of the greatest importance to him to know whether the same turbine will satisfactorily meet the requirements of a different specification.

For this purpose correction curves are employed, by means of which it is possible to determine, from the observed consumption of a turbine under actual test conditions, what its performance would be if either the vacuum, the initial pressure or the superheat were altered.

In this connection it is of great interest to note that superheat corrections are practically identical for every type of turbine, and over a wide range are nearly independent of the initial pressure. Further, the reduction in steam consumption by superheating is much greater than is theoretically due from the increased temperature of the steam. Apparently the benefit of superheat arises in the main from an accompanying reduction in the coefficients of steam friction, and hence a turbine with a high superheat will show a greater thermodynamic efficiency than a similar turbine supplied with steam initially dry and saturated. An increase in the vacuum also causes a reduction in the steam consumption per kilowatt, but in this case the gain is almost invariably less than that theoretically due. Moreover, the vacuum correction is not the same for all turbines. This is particularly the case where turbines have been built with a somewhat restricted steam way at the low-pressure end, such turbines being able to take less advantage of a good vacuum than others in which greater liberality has been shown in this regard.

The steam consumption also diminishes on raising the initial

pressure, but to a reasonable degree of approximation the correction required for this can be reduced to a vacuum correction, since the steam consumption per kilowatt-hour of a turbine taking steam initially at a pressure of 90 lb. per sq. in. absolute and discharging at a 28-in. vacuum will be very approximately the same as if the turbine took steam at 180 lb. per sq. in. absolute and exhausted against a 26-in. vacuum. In other words, if the initial and final pressures are increased in the same ratio, the efficiency of the turbine is almost unaltered, but its output is increased in the same ratio as the pressures. Hence in the case taken above, the pressures being doubled, the output would also be doubled. In fact, the quantity of steam which passes through a turbine is directly proportional to the absolute pressure below the governor valve. It may also be mentioned that, in the case of a reaction turbine, the velocity of inflow into the first row of blades is nearly constant at all loads, and is thus independent of the pressure below the governor valve, and, within very wide limits, is also independent of the vacuum in the condenser. (The weight passed is proportional to the density.) A set of corrections for high-pressure condensing turbines are given in Fig. 36.

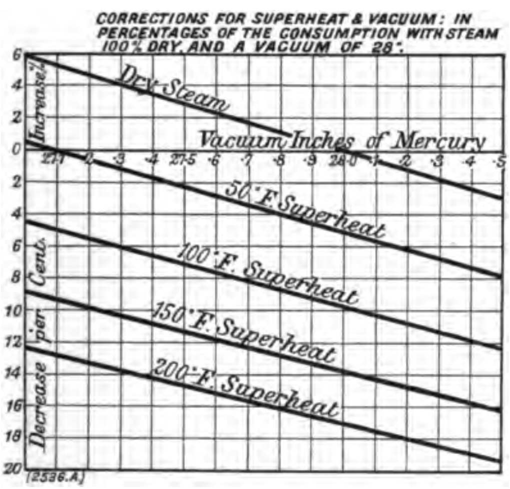


Fig. 36.

Thus, suppose that in a test the vacuum was $28\frac{1}{2}$ in. instead of 28 in., and the pressure below the governor valve was 120 lb. instead of 180 lb. absolute, whilst the superheat was 50 deg. instead of 150 deg. Then, if the consumption under the stated conditions was $16\frac{1}{2}$ lb. per kilowatt, the consumption under the specified conditions can be obtained from the correction curves.

Taking the pressure correction first, the consumption per kilowatt with 180 lb. absolute below the governor valve would be nearly the same as the observed consumption, provided that the back pressure were raised in the proportion of $\frac{180}{120}$. The actual vacuum was

$28\frac{1}{2}$ in., corresponding to a back pressure of $1\frac{1}{2}$ in. of mercury. Hence the equivalent vacuum with 180 lb. pressure will be $\frac{180 \times 1.5}{120} =$

Fig. 37. SUPERHEAT CORRECTIONS FOR HIGH-PRESSURE TURBINE DESIGNED FOR SUPERHEAT.

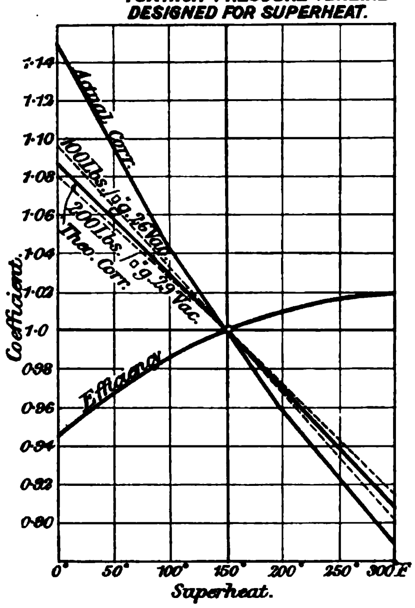


Fig. 39. VACUUM CORRECTIONS FOR HIGH-PRESSURE TURBINE DESIGNED FOR VACUUM.

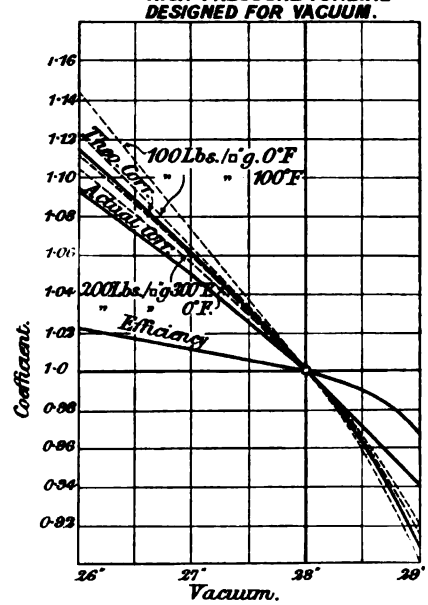


Fig. 38. SUPERHEAT CORRECTIONS FOR LOW-PRESSURE TURBINE DESIGNED FOR SUPERHEAT.

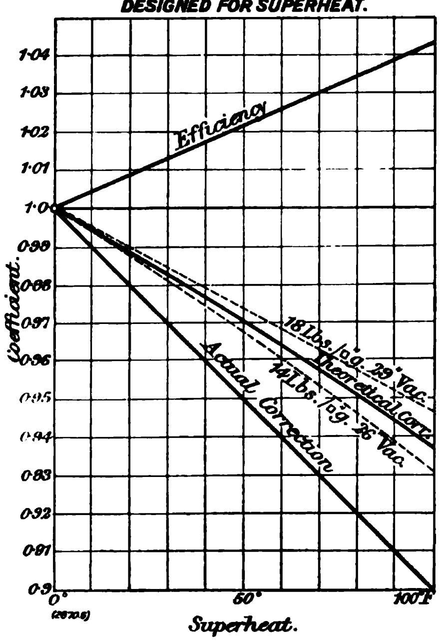
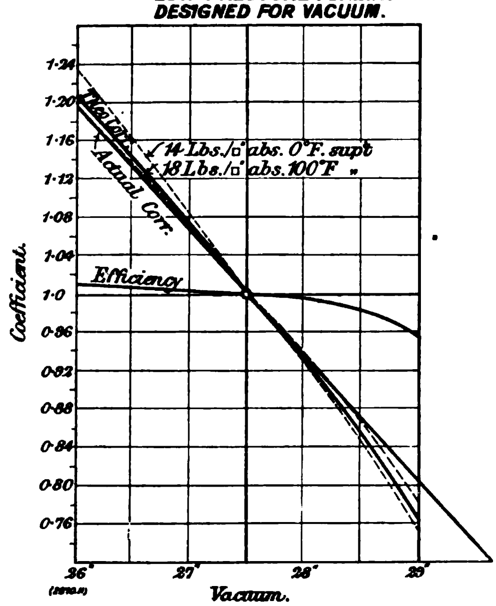


Fig. 40. VACUUM CORRECTIONS FOR LOW-PRESSURE TURBINE DESIGNED FOR VACUUM.



2.25 in., equivalent to a $27\frac{3}{4}$-in. vacuum. The consumption with such a vacuum will be, as stated, equal to that actually observed—viz.,

16.5 lb. per kilowatt. From the curves it appears that the same turbine, when working with no superheat but with a vacuum of 28 in., will take $3\frac{1}{2}$ per cent. more steam per kilowatt-hour, so that under these conditions the water rate would be 16.5×1.035 lb. per kilowatt-hour. If now, the other conditions remaining unchanged, the steam is given a superheat of 150 deg. Fahr., the water rate will be diminished by 13.8 per cent.; so that the consumption will be $\frac{16.5 \times 1.035}{1.138} = 15.0$ lb. per kilowatt-hour.

FIG. 41. PRESSURE CORRECTION FOR HIGH-PRESSURE TURBINE DESIGNED FOR THE PRESSURE.

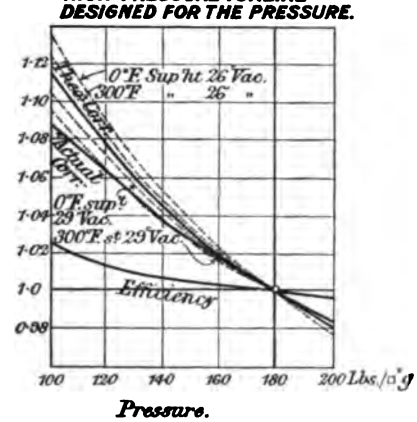
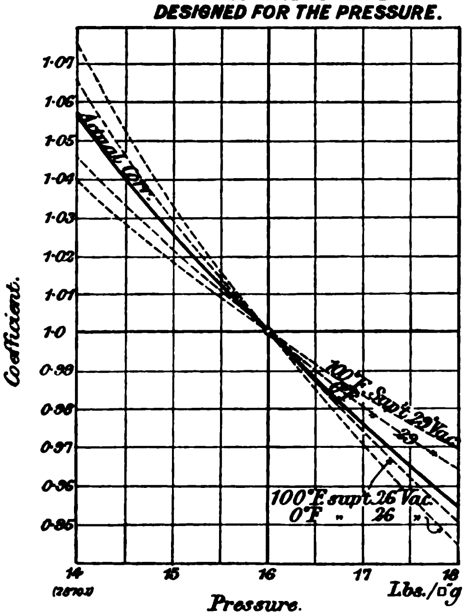


FIG. 42. PRESSURE CORRECTION FOR LOW-PRESSURE TURBINE DESIGNED FOR THE PRESSURE.



A very valuable series of correction curves was given by Mr. K. Baumann in a paper read before the Institution of Electrical Engineers at Manchester, in 1912, from which Figs. 37 to 42 are reproduced by permission of the Council.

Another useful curve is that of total steam consumption. This is always nearly a straight line, so that once two points on it are obtained, it is possible to determine the consumption at any load with very considerable accuracy. With a high-pressure turbine the total consumption at no load is generally approximately equal to the full load total consumption $\times \frac{15}{p}$, where p is the initial absolute pressure below the governor valve at full load. The amount varies, however, a little with different types of turbine.

CHAPTER VII.

THE PROVISIONAL PROPORTIONING OF A STEAM TURBINE.

EVEN where it is intended to make a thorough analysis of a proposed steam turbine, it is of advantage to have some simple method of quickly arriving at the general proportions necessary to attain a desired result. A number of rules permitting this to be done were drawn up some years since by Sir C. A. Parsons and his associates, but though originally devised for use with the reaction type of turbine, a similar method is equally applicable to all other varieties.

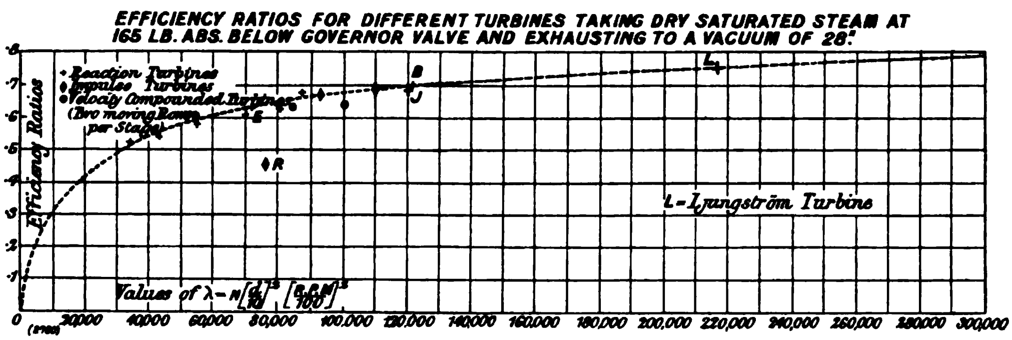


Fig. 43.

The curve given in Fig. 43 shows what efficiency may be expected from a *large* steam turbine when a certain coefficient λ is varied. This coefficient is given by the relation

$$\lambda = N\left[\frac{d}{10}\right]^2 \left[\frac{\text{R.P.M.}}{100}\right]^2,$$

where N denotes the total number of rows of moving blades, d the diameter in inches of the blade path, and R.P.M. the revolutions per minute. Thus a turbine consisting of nine wheels, 40 in. in diameter, run at 3000 revolutions per minute, will have the same efficiency as one consisting of eight wheels, 60 in. in diameter, run at 2120 revolutions per minute, since the value of λ is the same for both—

viz., 129,600. The efficiency given by the curve requires correction if the conditions are in any way abnormal. For example, in turbines of small output the losses by fan action, leakage, &c., are very high proportionately to the useful work done. Thus the point marked R on the diagram refers to a turbine of 150 kw. output, and, as will be seen, the efficiency is much below the normal. Again, if the attempt is made to combine a very large output with a high speed of revolution, there is (with a good vacuum) a difficulty in getting sufficient steam way at the last stage. The residual velocity of the steam as finally discharged may therefore be high, and the corresponding loss by "carry-over" may be a material fraction of the total available energy supplied to the turbine. Such cases require special treatment; but even then Fig. 43 is useful, as it still gives, with all necessary accuracy, efficiency ratios which are relatively correct.

Suppose, for example, that in an actual turbine the efficiency ratio obtained with the above value of λ is only 64 per cent. instead of 70.5 per cent. Let the by-pass be opened, short-circuiting one-sixth of the turbine, and thus reducing the effective value of λ to 108,000. From the curve the corresponding value of the efficiency ratio is about 69 per cent., then the actual efficiency ratio with the by-pass opened will be, approximately, $\frac{64 \times 69}{70.5} = 62.6$ per cent.

Under the standard conditions for which the diagram is drawn an ideal turbine would require 10.67 lb. of steam per kilowatt-hour, and if ϵ denote the efficiency ratio read from Fig. 43, the actual consumption per shaft kilowatt-hour will be, approximately, $\frac{10.67}{\epsilon}$. To get the consumption per kilowatt-hour measured at the switchboard, this figure must be divided by the generator efficiency.

The value chosen for λ is mainly fixed by commercial considerations. The higher the value of this coefficient the more costly the turbine, but this increase in cost is far from being proportional to the increase of the coefficient. The valves, governor, steam chest, &c., are the same, whatever the value of λ. Hence an increase of 50 per cent. in this coefficient generally means much less than a 50 per cent. increase in the cost of the turbine. The value to be adopted is, moreover, dependent on the design of the turbine. For a turbine

having two velocity stages per compartment, the value of λ should not exceed about 100,000 as a maximum, even in the case of large machines, unless both superheat and vacuum are abnormally high. With simple impulse turbines the value of λ may reach about 180,000 to 190,000 before the efficiency begins to diminish, and with the reaction type the limit is still higher. With given initial and final conditions, the choosing of a value for λ is, in effect, the same as fixing a value for the ratio $\frac{\text{blade speed}}{\text{steam speed}}$. As was shown in Chapter IV., for simple impulse turbines the best value for this ratio is commonly between 0.4 and 0.5, and, with the limits of steam pressure, superheat, and vacuum usual in power-station practice, a ratio of about 0.47 corresponds to a value of $\lambda = 190{,}000$. For reaction turbines, the ratio of blade speed to steam speed giving the highest *diagram* efficiency is about 0.9, which corresponds in ordinary power-station practice to a value of $\lambda = 350{,}000$ or so. So high a figure as this has never yet been reached, and, in view of the very flat character of the efficiency curve, it may be doubted whether the brake efficiency, as distinct from the indicated, would not reach its maximum value with a much lower value of λ. The Ljungström turbine, for which a point is plotted in Fig. 43, has the highest value of λ realised up to date; the ratio of blade speed to steam speed being rather over 0.7. With extremes of pressure and vacuum the above limits for λ may be a little raised, but it is not often, if ever, practicable to take *full* advantage of very high vacua.

The point on the curve, Fig. 43, marked B, was obtained with a 1500-kw. reaction turbine, but in this case, unquestionably, the clearances had been kept smaller than was desirable, and a strip occurred after some months running. The point marked J has reference to a machine rated at 6000 kw., whilst the point E corresponds to the famous Elberfeld turbine, and is a remarkable result to be obtained with so small a machine. The other points plotted for reaction turbines have been reduced from the progressive trials of a very large and very efficient marine turbine. The highest two points plotted from results obtained with impulse turbines refer to units, each rated at about 6000 kw. The point R, on the other hand, represents the results obtained with a 150-kw. unit of an early pattern

having rather large losses. The velocity-compounded machine with the higher value of λ was rated at about 4000 kw. The point with a coefficient of about 53,000 is a 400-kw. machine.

It should, perhaps, be noted here that all the points plotted for reaction turbines were obtained with machines having blades of the Parsons standard type. Where foundation rings are adopted, involving the use of blades of distorted forms, and the production of eddies by the projecting portions of the foundation rings, lower efficiencies must be expected.

By means of the correction curves given in the preceding Chapter, any conditions specified as to the initial pressure, superheat, or vacuum, can be reduced to the standard adopted for Fig. 43. Thus, supposing the data given are steam pressure 185 lb. absolute below the governor valve, superheat 150 deg. Fahr., and vacuum 28 in., the output of the turbine to be 2000 kw. measured at the switchboard, the generator efficiency being 95 per cent., and the revolutions 1500 or 3000 per minute, the general proportions of the turbine can still be got out as if the conditions were the same as those for which the curve in Fig. 43 has been drawn.

Suppose the turbine is, in the first instance, to be of the compartment-compounded impulse type, and that the speed adopted is 3000 revolutions per minute. A reference to Fig. 43 shows that with $\lambda = 100{,}000$ an efficiency ratio (reckoned at the turbine shaft) of about $67\frac{1}{2}$ per cent. may be expected. The coefficient is often higher for an impulse turbine of this output, but will serve for the purpose of illustration. We have then

$$N\left[\frac{d}{10}\right]^2\left[\frac{3000}{100}\right]^2 = 100{,}000,$$

so that

$$N\left(\frac{d}{10}\right)^2 = 111.1.$$

In order to minimise losses through excessive speed of the steam at final discharge to the exhaust, d^2 should not, with a 28-in. vacuum, be less, as a minimum, than $0.57 \times$ output in kilowatts, so that the least value of d^2 is 1140. Hence, with a speed of 3000 revolutions per minute, the maximum value of N desirable is about 10.

On the other hand, if convergent nozzles are to be used, the velocity of efflux must not exceed some 1500 ft. to 1700 ft. per second as a maximum, corresponding to a heat drop, per compart-

ment, of 45 to 58 B.Th.U. Allowing for the reheat factor, the total heat which becomes available whilst the steam expands from the initial to the final conditions will generally be somewhere about 360 to 380 B.Th.U. Hence the value of N cannot, so long as we are restricted to convergent nozzles, be less than 7. It may be noted in passing that it is this condition, rather than the centrifugal stresses, which may limit the size of a compartment-compounded turbine which can be built to run at 3000 revolutions per minute.

With a given coefficient λ, the fixing of the value of N, gives at once the value of d. The smaller N the larger d must be. As it is advantageous to reduce the number of separate parts and to shorten the turbine as much as possible, it is natural to take N at its minimum value, but in some cases this course is debarred by the difficulty or cost of obtaining large discs. This limitation, however, does not apply in the present instance, and hence, taking N as 7, we get

$$\left[\frac{d}{10}\right]^2 = \frac{111.1}{7} = 15.87,$$

so that d = 40 in. nearly. Hence the make up of the turbine would consist of seven wheels 40 in. in diameter. Its consumption per shaft kilowatt hour, under the conditions specified for Fig. 43, will be about $\frac{10.67}{0.675} = 15.80$ lb.

From the correction curves given in Chapter VI. it will appear that raising the initial pressure to 185 lb. will diminish this consumption by 2 per cent., whilst the superheat of 150 deg. Fahr. will give a further reduction of about 15 per cent. Hence the consumption per shaft kilowatt for the conditions originally specified will be somewhere about $\frac{15.80}{1.17} = 13.5$ lb. nearly.

Let us take now the case of a reaction turbine of the same output to be supplied with dry steam at 165 lb. per sq. in. below the governor valve, and to exhaust at 28 in. vacuum. Taking λ at 100,000, as before, the curve, Fig. 43, gives about $67\frac{1}{2}$ per cent. as the approximate efficiency ratio, so that the consumption per shaft kilowatt will be $\frac{10.67}{0.675} = 15.80$ lb. per hour, which could, of course, be reduced by superheating. Taking the output of the turbine as

2000 shaft kilowatts, the weight w of steam passing per second will be $\frac{2000 \times 15.80}{3600} = 9$ lb. per second nearly.

In the case of an impulse turbine there is, as already stated, a lower limit to the mean diameter d of the blading at the low-pressure end. Exactly the same limitation applies to reaction turbines, and d^2 should in no case be less than 0.57 × output in kilowatts. Where a very high vacuum is anticipated, d^2 is often made equal to the output in kilowatts, or in the case of a machine with a double-flow low-pressure end, d^2 may be 0.5 of the above values.

The reaction turbine is subject to another limit to the diameter of drum it is desirable to use, this limit applying to the high-pressure end. In the case of land turbines it is found advisable to limit the blade height to not less than $\frac{1}{25}$th of the drum diameter. If the blades are shorter than this the loss by tip leakage may become excessive. In marine practice the ratio of blade height to drum diameter may, however, be as little as $\frac{1}{75}$th, but there is, of course, a corresponding sacrifice of efficiency.

The above limit for land turbines leads to the rule that d_1, the drum diameter at the first row of blades, shall not be less than is given by the following equation:—

$$d_1^3 = \frac{1300\, w\, V_0 \sqrt{\lambda}}{\text{R.P.M.}}.$$

Here w has been found above to be 9 lb. per second, V_0 is the specific volume of the steam at the initial pressure of 165 lb. per sq. in., and R. P. M. stands for the revolutions made per minute.

Take R. P. M. as 1500 and we get

$$d_1^3 = \frac{1300}{1500} \,.\, 9 \,.\, \times 2.76 \times \sqrt{100{,}000},$$

whence $d_1 = 18.92$, which is the largest drum diameter it is desirable to use. If we take d_1 as 18 in., we can find the number of moving rows of blades by applying the formula

$$N \,.\, \left[\frac{d_1}{10}\right] \left[\frac{1500}{100}\right] = 100{,}000.$$

Thus N = 136, which is the number of moving rows which would be required if the drum were made 18 in. in diameter throughout. As already mentioned, however, the size of the low-pressure end must not be smaller than a certain limit. Hence it is usual

to construct the drum of a reaction turbine in three diameters, constituting a high-pressure, an intermediate, and a low-pressure section of the turbine. Of these the high-pressure and intermediate sections each do one-fourth of the total work, and the low-pressure section the rest.

The blading of the turbine is often divided up into twelve groups, the blade height in each group being made equal to that of the preceding multiplied by $\sqrt{2}$. (This is a purely empirical rule.) Hence the number N should be divisible by 12, which 136 is not, and N may therefore be increased to 144, slightly raising the value of λ. As the high-pressure end is to do one-fourth the total work, it will carry three groups, each comprising twelve rows of moving blades. The drum diameter for the intermediate section is made $\sqrt{2}$ times that of the high-pressure section, or, say, $25\frac{1}{2}$ in. It also carries three groups of blades, but the number per group (see page 58) is $\left(\frac{1}{\sqrt{2}}\right)^2 = \frac{1}{2}$, as many as on the high-pressure drum, thus there will be six instead of twelve moving rows per group. Finally, the low-pressure drum is made $\sqrt{2}$ times the diameter of the intermediate drum; doing twice the work, it has six in place of three groups of blades, but the diameter being double that of the high-pressure drum, there are only one-fourth the number of blades per group, that is, three instead of twelve.

Since the blade heights from group to group increase in the ratio of $\sqrt{2}$ to 1, the whole make up of the turbine is settled, once the height of the first row of blades is fixed.

Experience shows that the speed of inflow into the first row of a reaction turbine of ordinary design is given approximately by the semi-empirical relation $v_0 = \frac{2670}{\sqrt{N}}$, where N denotes the number of rows of moving blades.

$$\text{Hence } v_0 = \frac{2670}{\sqrt{144}} = 222.5 \text{ ft. per second.}$$

Some 7 to 8 per cent. of the total steam admitted to the turbine may be expected to pass through the high-pressure dummy, and thus the weight which actually flows through the high-pressure section of the turbine may be taken to be 8.39 lb. per second, a

quantity which may be denoted by w_1. Allowing for the clearance over the blade tips, the area, through the blading, available for flow may be taken as approximately one-third the annulus comprised between the drum and the casing. This area, which we may call Ω, is accordingly equal to

$$\frac{1}{3}\frac{\pi \cdot h \cdot (d_1 + h)}{144} = \frac{h \cdot (d_1 + h)}{137.5} \text{. sq. ft.}$$

The volume passing per second is $\Omega\, v_0 = w_1\, V_0$ where V_0 denotes the specific volume of the steam.

Hence

$$h = \frac{137.5 \cdot w_1\, V_0}{v_0\,(d_1 + h)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

Putting $w_1 = 8.39$, $V_0 = 2.757$; $v_0 = 222.5$, and $d_1 = 18$ in., we get

$$h = \frac{137.5 \times 8.39 \times 2.757}{222.5\,(18 + h)} = \frac{14.32}{18 + h},$$

h being small in comparison with 18 in., its value is most easily found by approximation.

Thus an approximate value of h is $\frac{14.32}{18} = 0.796$, a nearer value is now got by putting $h = \frac{14.32}{18 + 0.796}$, whence $h = 0.762$ in.

Electric-lighting turbines have usually to be capable of taking an overload, and a by-pass is accordingly provided round the first group. In order to maintain end balance, when the by-pass is in action, the mean diameter of the first group should be the same as that of the next, and it is usual, therefore, to swell up the drum at the first group (see description of Brush Turbine in a later Chapter). Hence in the present case the drum diameter at the first group may be increased to $18\frac{1}{4}$ in., and the make up of the high-pressure section of the turbine will then be as follows:—

Group number	1	2	3
Drum diameter, inches	$18\frac{1}{4}$	18	18
Number of moving rows per group... ...	12	12	12
Calculated blade height, inches	$\frac{3}{4}$	$\frac{3}{4} \times \sqrt{2}$	$\frac{3}{4} \times 2$
Actual blade height, inches	$\frac{3}{4}$	1	$1\frac{1}{2}$

If the high-pressure section had had four instead of three groups, so as to do one-third instead of one-fourth the total work,

this fourth group would have had a blade height of $\frac{3}{4} \times 2\sqrt{2} = 2$ in. nearly. Actually this group is transferred to the intermediate section, where a given length of blade will give approximately $\sqrt{2}$ times as much steam way. As at the same time there are only half as many blades per group, the velocity of the steam will be $\sqrt{2}$ as great as in the case of the high-pressure section. Hence a given blade height on the intermediate section will pass twice as much steam as it would on the high-pressure section, and since a fourth group on the latter would have a blade height of 2 in., the group which replaces this should have a blade height of $2 \times \frac{1}{2} = 1$ in. Hence the make up of the intermediate section will be :—

Group number	4	5	6
Drum diameter, inches	$25\frac{1}{2}$	$25\frac{1}{2}$	$25\frac{1}{2}$
Number of moving blades per group ...	6	6	6
Calculated blade height, inches	1	$1 \times \sqrt{2}$	1×2
Actual blade height, inches	1	$1\frac{1}{2}$	2

Blade heights are taken to the nearest $\frac{1}{8}$ in., as it is found that there is no material advantage in adjusting them more finely.

In an exactly similar way the height of the first group on the low-pressure section is one half that of a hypothetical fourth group on the intermediate section which it replaces. The latter would have a height equal to 3 in., and thus the make-up of the blading for the low-pressure section will be as follows :—

Group number	7	8	9	10	11	12
Drum diameter, inches	36	36	36	36	36	36
Number of moving rows per group ...	3	3	3	3	3	3
Blade height, inches	$1\frac{1}{2}$	$2\frac{1}{8}$	3	$4\frac{1}{4}$	6	6*

* Semi-wing and wing blades.

The above method of arriving at the general proportions of a reaction turbine to give a specified output and steam consumption it will be seen is exceedingly simple, but in spite of that, so long as standard practice is followed, it gives results on which it is difficult to improve much by the most elaborate analysis.

The semi-wing blade used in Group 12 is merely a normal blade, set so that the steam way between a pair of adjacent blades

is half the pitch. Its use could be avoided by making the drum in four diameters instead of three. A detailed study of the work done in each group of the above turbine is given in a succeeding Chapter.

The method of proportioning above described, can also be adopted for proportioning marine turbines, and is exceedingly convenient in getting out and comparing preliminary designs. Generally, however, the clearances over blade tips are proportionately considerably larger than in land practice, and this reduces the efficiency. In the example above taken the blading was divided up into twelve groups, the blade height from group to group increasing in the ratio of $\sqrt{2}$ to 1. An alternative method also largely adopted is to divide up the blading into sixteen groups, increasing in the ratio of $\sqrt[3]{2}$ to 1. In cases as few as eight groups have been adopted, the blade height ratio being then made equal to $\sqrt[3]{4}$, but this involves some sacrifice of efficiency.

It is quite possible to apply the curves of Fig. 43 for getting out the general proportions of what is known as a disc-and-drum machine. In such machines about one-third the total work is accomplished in the first stage, which is fitted with a velocity-compounded wheel having two rows of blades, whilst the rest of the turbine is constructed on the reaction principle, and does two-thirds of the total work.

Suppose, as before, that λ is 100,000, but that the speed is to be 3000 revolutions per minute. If the machine were wholly built on the velocity-compounded system, it would consist of three compartments, each having a wheel with two rows of moving buckets, so that N = 6. Hence we get

$$6 \times \left[\frac{d}{10}\right]^2 \cdot \left[\frac{\text{R. P. M.}}{100}\right] = 100{,}000.$$

Whence

$$\left(\frac{d}{10}\right)^2 = \frac{100{,}000}{6 \times 900} = 18.52,$$

or d = 43 in. nearly.

Thus the high-pressure end of the proposed turbine would consist of one velocity-compounded wheel, 43 in. in mean diameter.

Coming to the reaction end, it has to be noted that with a large output and a speed of 3000 revolutions per minute, the diameter of the low-pressure end becomes the controlling factor. The mean

diameter d of the last row of blading must not be less than is given by the relation $d^2 = 0.57 \times$ output in kilowatts $= 1140$.

Whence $d = 34$ in. nearly. The largest blade length that can be used is one-fifth the mean diameter, so that the drum diameter may be taken to be 27 in.

To find the number of moving rows in the turbine, if it were wholly a reaction machine, we have

$$N \cdot \left[\frac{d}{10}\right]^2 \left[\frac{\text{R. P. M.}}{100}\right]^2 = 100{,}000,$$

or

$$N = \frac{100{,}000}{900 \times 2.7} = 15.$$

Fifteen rows will not divide up into twelve groups, but sixteen will into sixteen, there being then one moving row in each group. Hence N. may be taken as sixteen instead of fifteen, and the blade heights will then increase in the ratio of $\sqrt[3]{2}$ to 1.

The coefficient λ being nearly the same as before, the weight of steam passed per second will also be nearly the same, viz., 9 lb. per second. There is, however, now no high-pressure dummy, so that the weight actually passing through the blading may be taken as 8.82 lb. per second, the residue passing direct to the exhaust through the balancing dummy.

Considering the turbine still to be wholly a reaction one, the velocity v_0 of steam into the first row of blades will be, approximately,

$$\frac{2670}{\sqrt{16}} = 667.5 \text{ ft. per second.}$$

From the equation (8) we have for the theoretical blade height here

$$h = \frac{137.5\, w_1\, V_0}{v_0\,(d + h)} = \frac{137.5 \times 8.82 \times 2.757}{667.5\,(d + h)}.$$

$$\therefore h = \frac{4.94}{27 + h}$$

$$\text{or } h = 0.182, \text{ say } \tfrac{3}{16} \text{ in.}$$

This is, of course, merely what the blade height would have been at the first row if the whole turbine were a reaction drum with sixteen rows of moving blades. Actually, however, the first five rows will be replaced by the velocity-compounded wheel, but from the value of h found we can get the blade heights throughout, simply by multiplying in succession by $\sqrt[3]{2} = 1.259$.

We thus get the following table:—

Group Number.	Blade Height.	
	Calculated.	Actual.
	inches.	inches.
1	$\frac{3}{16}$	—
2	0.246	—
3	0.297	—
4	$\frac{3}{8}$	—
5	0.472	—
6	0.596	$\frac{5}{8}$
7	$\frac{3}{4}$	$\frac{3}{4}$
8	0.945	$\frac{15}{16}$
9	1.189	$1\frac{3}{16}$
10	$1\frac{1}{2}$	$1\frac{1}{2}$
11	1.887	$1\frac{7}{8}$
12	2.38	$2\frac{3}{8}$
13	3	3
14	3.78	$3\frac{3}{4}$
15	4.77	$4\frac{3}{4}$
16	6	6

The reaction part of the turbine will thus commence at Stage 6. The tip speed at the last row of blades is 510 ft. per second, which is nearly the highest yet used with reaction blading. A blade height of $\frac{5}{8}$ in. on a drum 27 in. in diameter is rather small, but still smaller proportions of blade height to diameter are common in marine practice. An alternative in the present case would be to transfer Groups Nos. 6, 7, and 8 to a smaller diameter of drum. If the latter be made equal to $27 \div \sqrt{2} = 19$ in., then each of these groups will contain two rows of moving blades instead of one. By this transference the blade height for Group 8 will be doubled, since the mean circumference is decreased approximately in the ratio of 1 to $\sqrt{2}$, and at the same time the number of rows per group being doubled, the steam speed will be diminished also in the ratio of 1 to $\sqrt{2}$. Hence this group, after transference to

the smaller diameter, will have a calculated blade height equal to $0.945 \times 2 = 1.890$ in.

The make-up of the 19-in. section of the drum will now be :—

Group number	6	7	8
Number of moving rows per group...	2	2	2
Calculated blade height, inches	1.192	1.51	1.890
Actual blade height, inches	$1\frac{3}{16}$	$1\frac{1}{2}$	$1\frac{7}{8}$

Fig. 43 can also be used for getting out the general proportions of exhaust-steam turbines, the number of rows given by the formula

$$N \cdot \left(\frac{d}{10}\right)^2 \left(\frac{\text{R. P. M.}}{100}\right)^2$$

being halved as exhaust steam has less energy per pound.

In the foregoing, three alternative designs for turbines of the same output have had their principal dimensions fixed by simple practical rules. In succeeding Chapters methods of analysing them in detail from the standpoint of hydraulics and thermodynamics will be discussed.

In the case of marine turbines it is often necessary to use a low coefficient, so as to save weight and length. At the same time, and for the same reason, the diameter at the high-pressure end has to be made much greater than is usual with land turbines, and this implies a considerable loss by leakage over blade tips. The area of the annulus between the casing and the drum at the low-pressure end should not be less than about 2 sq. ft. per 1000 shaft horse-power, but there is generally no difficulty in satisfying this limit.

An empirical rule for the drum diameter in inches at the high-pressure end is

$$d = 2.25 \sqrt[3]{\text{shaft horse-power}},$$

where "shaft horse-power" denotes the total horse-power developed in the whole series of turbines by that steam which has passed into the one high-pressure turbine.

This rule gives results in fair agreement with general practice. A smaller value of d would reduce the loss by leakage over blade tips, but would increase the length of the turbine. The low-pressure drum is made equal to $d\sqrt{2}$. On the three-shaft arrange-

ment the high-pressure turbine is designed to develop about one-third the total work, and carries, therefore, one-third the total number of groups. With the four-shaft arrangement each high-pressure turbine exhausts into one low-pressure turbine only, and the number of groups is generally, though not always, equally divided between the two.

Thus, suppose it is required to develop 8000 shaft horse-power at 465 revolutions per minute, with a coefficient of about 65,000, then

$$d = 2.25 \sqrt{8000} = 45 \text{ in.}$$

We have

$$N \cdot \left(\frac{45}{10}\right)^2 \left(\frac{465}{100}\right)^2 = 65{,}000.$$

Whence N = 148. If there are, as usual, twelve groups, N should be a multiple of 12, or say N = 144.

From Fig. 43 the efficiency will be about 0.61 that of a perfect turbine. The latter, in the conditions for which Fig. 43 is drawn, would require 8 lb. per brake horse-power per hour. Hence the actual steam consumption will be, approximately, $\frac{8}{0.61} = 13.1$ lb. per brake horse-power per hour for an initial pressure of 165 absolute. If the initial pressure be 175 absolute, the consumption will be approximately about 13 lb. per shaft horse-power per hour, which is equivalent to a weight of about 29 lb. per second. This figure will really be exceeded, owing to the large drum diameter leading to large losses by leakage over blade tips. To allow for this it may be assumed that this is not the total weight of steam but that which passes through the high-pressure blading, in addition to which there is, of course, a flow through the high-pressure dummy. Hence from equation (8) we get the blade height at the first group

$$h = \frac{137.5 \times 29 \times 2.61}{222.5\,(d + h)} = \frac{46.76}{45 + h}$$

whence $h = 1$ in. very nearly.

It is usual, in such an installation, for the turbine equipment to consist of one high-pressure turbine on a centre shaft discharging into two low-pressure turbines on wing shafts, the former having four groups of blades with heights increasing from group to group in the ratio of $\sqrt{2}$ to 1; whilst the low-pressure turbines each consist of eight groups with the same height ratio (apart from the wing blades), and in this case the blade height at the entrance to

each of the two low-pressure turbines is the same as it is for the first group of the high-pressure turbine, since the drum diameters are also increased in the ratio of $\sqrt{2}$ to 1.

The blading of the high-pressure turbine will thus consist of four groups of twenty-four rows each (half fixed and half moving), with blade heights as follow:—

Group number	1	2	3	4
Blade height, in inches	1	$1\frac{3}{8}$	2	$2\frac{3}{4}$

The low-pressure turbines will have a drum diameter equal to 45 in. $\times$ 1.41 = 63.6 in., say 64 in., and would consist of eight groups of twelve rows each (including both fixed and moving blades), since the number of rows in equivalent groups should be inversely proportional to the square of the blade speeds.

The make-up of each of the two low-pressure turbines would thus be as follows:—

Group number	1	2	3	4	5	6	7	8
Blade height, inches ...	1	$1\frac{3}{8}$	2	$2\frac{3}{4}$	4	$5\frac{1}{2}$	$5\frac{1}{2}$	$5\frac{1}{2}$

The last two groups would, of course, consist of semi-wing and wing blades.

CHAPTER VIII.

THE DESIGN OF AN IMPULSE TURBINE.

IT was shown in the preceding Chapter that an impulse turbine of 2000 kw. output and consisting of seven wheels, each 40 in. in mean diameter, running at 3000 revolutions per minute, would, with an initial steam pressure of 185 lb. per sq. in. absolute and with 150 deg. superheat (below the governor-valve) and a vacuum of 28 in., have a steam consumption of approximately 13.5 lb. per kilowatt-hour measured at the turbine shaft.

It now remains to fix the proportions of the blading.

With a 28-in. vacuum the area of the complete annulus at the last row of the blades should not be less than about 2.5 sq. ft. per 1000 kw., and in many cases as much as 4 sq. ft. is allowed, which makes it possible to utilise efficiently still higher vacua. The rule that 2.5 sq. ft. should be allowed per 1000 kw. is equivalent to that already given in a previous Chapter, viz., that d^2 should be greater than 0.57 × output in kilowatts, and that the length of blade must not exceed $\frac{d}{5}$.

From the Mollier diagram, Fig. 7, *ante*, we find that, for the assumed conditions, the available heat u = 359 B.Th.U., so that the theoretical consumption is $\frac{3412}{359}$ = 9.504 lb. per shaft-kilowatt hour. The data from which Fig. 43, page 50, was plotted were derived in the main from turbines of large size, in which the losses by gland leakage, bearing friction, &c., were probably of the order of 3 per cent. Thus the steam rate of the proposed turbine per "indicated" kilowatt, if of the same "mechanical" efficiency, will be about 13.5 × 0.97 = 13.10 lb. per indicated kilowatt-hour. Hence the indicated efficiency ratio is $\frac{9.504}{13.10}$ = 72.6 per cent., which, from the table of reheat factors, given on page 45, will correspond to an hydraulic efficiency of $\frac{72.6}{1.056}$ = 68.7 per cent.,

and the total heat units which become available as the steam passes through the turbine will thus be $359 \times 1.056 = 382$ B.Th.U., equivalent to a heat drop per compartment of $\frac{382}{7} = 54.6$ B.Th.U. Hence the theoretical velocity of efflux will be about 1655 ft. per second. This is, of course, above the critical value for non-superheated steam, but Rateau has shown by a series of direct experiments that a slight amount of under expansion in a guide blade is, if anything, beneficial to bucket efficiency, though a large amount of under expansion is bad. His results, which were obtained by directing a jet of steam into a Pelton-type bucket, on which the consequent thrust could be measured, are plotted in Fig. 44. In the case of curve A, at all points to the left of M, the discharge end of the nozzle was too large, and there was a great loss of efficiency. At all points to the right of M, on the other hand, the discharge area was too small, with consequent under expansion. It will be seen that this under expansion has to be large before there is any serious diminution in the efficiency. The curve B was obtained with another nozzle. In this case the discharge end was too large for all points to the left of N.

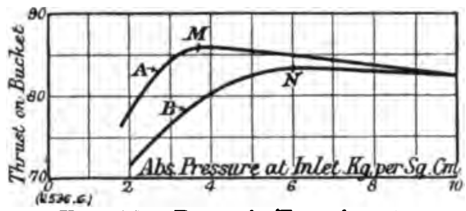


Fig. 44. Rateau's Experiments.

From Christlein's curves, Fig. 13, the actual velocity of efflux will be about 0.94 of the theoretical, so that the guide-blade efficiency is about 0.88. The bucket speed is 523.6 ft. per second, and the actual steam speed is $0.94 \times 1655 = 1552$ ft. per second, so that the ratio of blade speed to steam speed is $\frac{523.6}{1552} = 0.338$. From the tables, page 35, *ante*, this, with a bucket angle at discharge of 25 deg., will correspond to a bucket efficiency of about 0.717. Hence the stage efficiency is $0.88 \times 0.715 = 0.63$. This, however, corresponds to the condition of non-superheated steam and from the curve in Fig. 37, 150 deg. superheat will increase the average stage efficiency by 5.3 per cent. Hence the stage efficiency found in this way is 66.6 per cent. as against the 68.7 per cent. deduced from Fig. 43.

Where a curve such as Fig. 43 is plotted entirely from results of tests of a series of turbines of a particular type, the stage

efficiency deduced from this curve by means of the reheat factor will be more accurate than when it is obtained by estimating the turbine efficiency from the assumed efficiencies of the nozzles and buckets. Data as to turbine performances are, in fact, generally more readily obtainable than results of really trustworthy experiments on guide blades and bucket friction. In the present case, however, Fig. 43 has been plotted mostly from results obtained with reaction turbines, those from other types being fewer in number, so that the lower value found for the stage efficiency is as likely to be accurate as the higher. Taking a mean value gives $\eta = \dfrac{66.6 + 68.7}{2} = 67.6$. The corresponding "indicated" efficiency ratio is from the curves, Fig. 35, about 71.8. The corrected consumption per indicated kilowatt-hour will be about 13.25 lb. A 2000-kw. unit of the compartment-compounded impulse type will have rather higher internal losses than the larger turbines, from which Fig. 43 was derived, say 4 per cent. instead of 3 per cent., so that the consumption per shaft kilowatt-hour, as finally corrected, will be about $\dfrac{13.25}{0.96} = 13.8$ lb.

It remains to settle the proportions of the guide blades, and for that we need the total consumption of the turbine per hour. If the 2000 kw. is to be obtained at the switchboard more will be required at the shaft. Taking the generator efficiency as 95 per cent., the shaft kilowatts will be 2100, or say 2250, so as to allow a margin above the rated load. The total weight of steam passing the turbine is then $2250 \times 13.8 = 31{,}050$ lb. per hour or 8.625, say 8.63 lb. per second. The steam in its initial state has a pressure of 185 lb. absolute and a specific volume of 2.770 cub. ft. per lb.

The volume of the steam at exhaust can easily be found from the Mollier diagram in the manner explained in Chapter V., since we know the indicated efficiency ratio. We thus find that the dryness fraction at exhaust is about 91 per cent., giving a specific volume of $340 \times 0.91 = 309.4$ cub. ft. per lb. Strictly speaking, this is the volume of the steam as it leaves the buckets, the friction in which reheats it a little. This latter fact could readily be taken into account, but such a refinement is unnecessary. Hence we may take it that the last row of guide blades must be equal to passing 309.4×8.63

cub. ft. of steam at a speed of 1552 ft. per second. Hence if Ω be the area through these guide blades in square feet we have

$$1552 \cdot \Omega = 309.4 \times 8.63,$$

or

$$\Omega = 1.7206 \text{ sq. ft.}$$

Then if d denote the mean diameter in inches, h the length of the guide blades, α the angle of discharge, and F the thickness factor (see page 22)

$$\Omega = \frac{F \cdot \pi \cdot d \cdot h \cdot \sin \alpha}{144}.$$

Taking F as 0.93, d as 40 in., and α as 20 deg., we get

$$h = \frac{144 \times 1.7206}{0.93 \times \pi \times 40 \times 0.3420} = 6.200 \text{ in.}$$

The height required at the other stages is proportional to the specific volume of the steam there.

With only seven compartments it is quite possible to work out from the Mollier diagram the pressure and volume of the steam in each compartment step by step, as explained on page 43, Chapter V., *ante*; but an alternative method will be used here, as it will also be applicable however numerous the stages.

It is based on the closely approximate relation $p_0 V_0^\gamma = p V^\gamma$. In this case $p_0 = 185$, $p_x = 0.980$ lb. absolute, $V_0 = 2.770$, and $V_x = 309.4$ cub. ft. per lb. Hence

$$\gamma = \frac{\log p_0 - \log p_x}{\log V_x - \log V_0} = 1.1114.$$

If V_n denote the specific volume in compartment n, and p_n the pressure there, we have $V_n = \rho_n \cdot V_0$ and $p_n = \frac{p_0}{x_n}$.

Putting $A_n = \frac{\rho_n}{x_n}$, it is easy to show,* if the same amount of

---

* If the energy accounted for in each stage is to be the same we must have

$$144 \frac{\gamma}{\gamma - 1} (p_{n-1} V_{n-1} - p_n V_n) = \text{constant}.$$

Hence $\quad p_0 V_0 - p_1 V_1 = p_1 V_1 - p_2 V_2$

$$= p_2 V_2 - p_3 V_3, \text{ and so on.}$$

But $\quad p_1 V_1 = A_1 p_0 V_0 ; \; p_2 V_2 = A_2 p_0 V_0 ;$

and $\quad p_3 V_3 = A_3 p_0 V_0$; and so on.

Thus we get $\quad p_0 V_0 (1 - A_1) = p_0 V_0 (A_1 - A_2),$

so that $\quad 2 A_1 - 1 = A_2.$

Similarly $\quad p_0 V_0 (1 - A_1) = p_0 V_0 (A_2 - A_3)$

which gives $\quad 3 A_1 - 2 = A_3$

and generally $\quad n A_1 - n + 1 = A_n.$ Q. E. D.

work is done in each compartment, that

$$A_n = n\,A_1 - n + 1.$$

We have in the present case

$$A_n = \frac{\rho_n}{x_n} = \frac{309.4}{2.770} \times \frac{0.980}{185} = 0.59170.$$

Hence

$$7\,A_1 - 6 = 0.59170$$

and

$$\begin{aligned} A_1 &= 0.94167 \\ A_2 &= 2\,A_1 - 1 = 0.88334 \\ A_3 &= 3\,A_1 - 2 = 0.82501 \\ A_4 &= 4\,A_1 - 3 = 0.76668 \\ A_5 &= 5\,A_1 - 4 = 0.70836 \\ A_6 &= 6\,A_1 - 5 = 0.65003. \end{aligned}$$

Since

$$\rho^{\gamma} = x,\ A = \frac{\rho}{x} = \left(\frac{1}{x}\right)^{\gamma-1}, \text{ whence } \rho = \left[\frac{1}{A}\right]^{\frac{1}{\gamma-1}} \text{ or } \log \rho = \frac{1}{0.1114} \cdot \log \frac{1}{A}.$$

The calculation then proceeds as shown in Table IV.

TABLE IV.—THEORETICAL BLADE HEIGHTS FOR IMPULSE TURBINE.

Stage Number.	1	2	3	4	5	6	7
Log A	$\bar{1}.9739$	$\bar{1}.9461$	$\bar{1}.9165$	$\bar{1}.8846$	$\bar{1}.8503$	$\bar{1}.8129$	—
Log $\frac{1}{A}$	0.0261	0.0539	0.0835	0.1154	0.1497	0.1871	—
Log $\rho = \frac{1}{0.1114} \log \frac{1}{A}$	0.2343	0.4839	0.7496	1.0360	1.3440	1.6796	—
Log V = log ρ + log V_0	0.6768	0.9264	1.1920	1.4785	1.7865	2.1221	—
V ... cub. ft. per lb.	4.751	8.441	15.56	30.09	61.16	132.5	309.2
Theoretical blade height $= h_7 \times \frac{V}{V_7}$ ... in.	0.09526	0.1693	0.3120	0.6033	1.226	2.005	6.200

It will be obvious that in the first three stages the wheels must work with partial admission so as to avoid absurdly small blade heights. If, therefore, the blade height in these three compartments is taken as ½ in., then the segment covered by the guide blades will be 0.19 of the complete circumference in the case of Stage 1, 0.34 of the circumference in Stage 2, and 0.624 of the circumference in Stage 3. A smaller discharge angle than 20 deg. is often adopted in these partial admission stages. Thus, if the

discharge angle be made 16 deg. instead of 20 deg., the proportion of the whole circumference covered will be increased in the ratio of $\frac{\sin 20 \text{ deg.}}{\sin 16 \text{ deg.}}$.

For the last four compartments the blades may be taken to the nearest $\frac{1}{32}$nd, though this is a finer adjustment than is really necessary, a considerable departure from the calculated values having but little effect on the efficiency.

For some purposes it is desirable to know the pressure in each compartment. This is now easily found, since we have $A = \frac{\rho}{x}$ or $x = \frac{\rho}{A}$, and ρ and A are both known. Hence we get—

Stage number	1	2	3	4	5	6	7
$p = \frac{p_0}{x}$ lb. per sq. in. ...	101.6	53.75	27.27	13.13	5.974	2.534	0.980

In order to keep down disc friction and minimise fan action the clearance between the wheels and the diaphragms should be reduced to a minimum. It is also a good plan to shape the entrance to the guide blades so as to receive, at a favourable angle, the steam discharged from the preceding wheel. In this way some 40 to 50 per cent. of the "carry-over" from the wheel is usefully utilised in promoting flow through the guide blades. For an example of this method of shaping the guide blades, the drawings of the Westinghouse-Rateau turbine, reproduced in another Chapter, may be referred to.

It is particularly desirable to adopt this procedure when, as is often the case, the total supply of energy is unequally divided between the several stages. A particularly large "heat drop" is often allowed in Stage No. 1, so as to keep down the pressure in the first compartment, thus minimising leakage losses and disc friction, which are directly proportional to the density of the steam.

Suppose for example that the pressure in the first stage is 75 lb. instead of 101.6 lb., then, from the Mollier diagram, the heat accounted for in this compartment will be 81 B.Th.U., leaving 294 B.Th.U. to be equally divided amongst the remainder. The theoretical velocity of efflux at the first stage would be 2016, which would be too large for a non-divergent nozzle unless the steam

had its full superheat. The ratio of blade speed to steam speed here will, of course, be reduced to an uneconomical value, but a certain proportion of the loss by carry-over can be recuperated by suitably shaping the guide blades in the first diaphragm, so that the over-all efficiency will be much the same as before. Hence the relation of pressure and volume may be taken as before, viz., $p\,V^{1.1114}$ = constant. To determine the values of V in compartments 2, 3, 4, &c., we find V_1 from the above equation, and we then have

$$6\,A_2 - 5 = \frac{p_x}{p_1} \cdot \frac{V_x}{V_1}.$$

$$A_3 = 2\,A_2 - 1\ .\ A_n = 3\,A_2 - 2,$$

and so on, from which the corresponding values of ρ_2, ρ_3, &c., can be determined as before.

Referring back to the case in which the energy is equally shared amongst all seven compartments, suppose a by-pass is fitted to take an overload by short circuiting the first wheel. The weight of steam passed will be increased in the same ratio as the pressure in this first compartment, that is to say, in the ratio $\frac{185}{101.6}$. At the same time, however, the effective value of the coefficient λ will be diminished from 100,000 to about 86,000. Hence, from Fig. 43, page 50, the work done per pound will be diminished in the ratio of $\frac{66}{67.5}$. The net result is that the overload capacity will be increased to $\frac{185}{101.6} \cdot \times \frac{66}{67.5} \times 2000 = 3560$ kilowatts. This is more than the generator would carry, but is to some extent an over-estimate, since the vacuum would fall off, and the steam, owing to wire-drawing, would be rather less in pressure than the full 185 lb. in the first compartment.

It will be obvious that with a small total number of stages an overload will be best taken by opening up additional nozzles at the high-pressure end rather than by by-passing the steam.

CHAPTER IX.

VELOCITY-COMPOUNDED IMPULSE WHEELS.

FROM Fig. 30, page 31, *ante*, it will be evident that the slower the bucket speed the greater the value of r, and the greater the kinetic energy carried over. Hence when it is desired to combine a high velocity of efflux from the nozzle with a low bucket speed, the so-called plan of velocity compounding is frequently resorted to. This is represented diagrammatically in Fig. 45. Here A represents a converging-diverging nozzle, in which the steam is expended down from, say, 170 lb. per sq. in. to about atmospheric pressure, or a little below, and thus enters the wheel chamber at a high velocity, which may be 2560 ft. per second or so. This jet of rapidly-moving steam meets first the moving buckets B, which are mounted on a wheel keyed to the turbine shaft, and after giving up some of its kinetic energy to these buckets the steam is delivered from them still at a high speed, and meets next a set of buckets C fixed to the casing of the chamber. These divert the path of the steam jet, and direct it into a second set of moving buckets D, which, after abstracting from the steam a further proportion of its kinetic energy, pass it on to a second set of fixed buckets E, and these, again, on to a third moving wheel F, from which it is delivered with a very small remaining kinetic energy. There are two methods of drawing the diagram of velocities for such a turbine. One method is represented in Fig. 46, and this is probably the easiest to understand, though less convenient in practical use than a second method to be described later.

Here A B represents the velocity of issue, from the nozzle, taken as 2562 ft. per second, and the bucket speed is taken as 400 ft. per second. The line B C represents the velocity of the steam relatively to the bucket at entrance, and is found by scaling to be 2189 ft. per second, the bucket velocity C A being 400 ft.

per second. The angle of discharge in the case of the moving buckets is the same as that at entrance, and hence, considered relatively to this bucket, the steam flows out along the line C D. For simplicity, we shall, in this instance, assume a constant value of ψ equal to 0.885 throughout. Hence C D is made equal to 0.885×2189 ft., or 1937 ft. per second. By setting off D E, the bucket speed, we get C E = 1580 ft. per second as the absolute velocity of the steam at issue from the first row of buckets, and this is also the velocity with which it enters the first set of fixed buckets. In these buckets it is turned round, so that it discharges

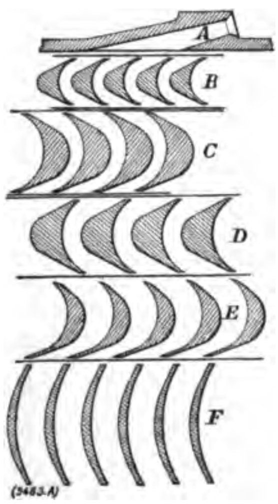


Fig. 45. Velocity Compounding.

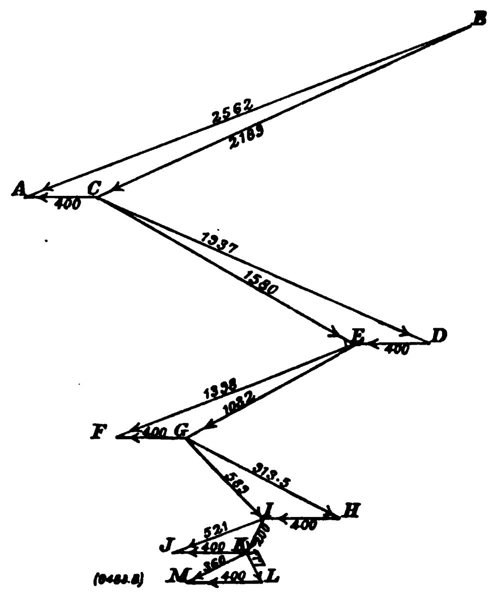


Fig. 46. Diagram of Velocities.

along the line E F, which makes an angle of 20 deg. with the plane of the wheel carrying the second set of moving buckets. It again loses in these fixed buckets 11.5 per cent. of its velocity, so that E F is 1398 ft. Setting off G F as shown, we get E G = 1032 ft., the velocity of the steam relatively to the second set of moving buckets. Multiplying E G by ψ we get G H = 913.5 ft., and by setting off H I = 400 ft. we find G I the velocity at entrance to the second set of fixed buckets to be 589 ft. per second, and from these it issues along I J at a speed of 521 ft. per second. Making K J = 400 ft., there results I K = 200 ft. per second as the velocity at entrance to the third set of moving buckets.

K L = 177 ft. is then the velocity at discharge, measured relatively to the moving wheel, which gives K M = 360 ft. per second as the absolute velocity of the steam at final discharge.

A loss of 11.5 per cent. in velocity corresponds to a loss of 21.85 per cent. in kinetic energy. Hence we have :—

Loss in first moving bucket =

$$0.2168 \times \left(\frac{2189}{224}\right)^2 = 20.70 \text{ B.Th.U.}$$

Loss in first fixed bucket =

$$0.2168 \times \left(\frac{1580}{224}\right)^2 = 10.79 \quad ,,$$

Loss in second moving bucket =

$$0.2168 \times \left(\frac{1032}{224}\right)^2 = 4.60 \quad ,,$$

Loss in second fixed bucket =

$$0.2168 \times \left(\frac{589}{224}\right)^2 = 1.50 \quad ,,$$

Loss in third moving bucket =

$$0.2168 \times \left(\frac{200}{224}\right)^2 = 0.17 \quad ,,$$

Carried away as kinetic energy =

$$\left(\frac{360}{224}\right)^2 = 2.58 \quad ,,$$

Total losses = 40.34 ,,

Now the original kinetic energy in the steam was $\left(\frac{2562}{224}\right)^2 =$ 130.8 B.Th.U. per lb., so that *e*, the diagram efficiency, is equal to $\frac{130.8 - 40.34}{130.8} = 0.6916$. This is a considerably higher figure than would be realised in actual practice, since the value of ψ has been taken too high.

The method of determining *e* above given has the advantage of clearness, but is not nearly so convenient in practice as another based on Fig. 31, Chapter IV., *ante*, in which this efficiency is determined from the "impulse" on the wheel.

The diagram used in this case is represented in Fig. 47, which has been drawn for two sets of moving buckets. Corresponding points are lettered similarly to Fig. 31. Hence as before A B denotes the velocity of the steam as originally delivered, B C the speed of the bucket, and E C the relative speed with which the steam is finally delivered from the first row of buckets.

The impulse in this row, per pound of steam passed, is as before

$$\frac{1}{g} \text{F G}.$$

E C being the relative velocity of the steam on discharge from the first row, its absolute velocity is equal to E B, and with this it enters the row of fixed buckets. In this it is turned round and discharged in the direction B K and with a velocity such that B K = ψ . E B.

This row of fixed buckets stands to the second row of moving buckets in the same relation as the guide blades did to the first row. Hence we can find the impulse on this second row of moving buckets by exactly the same process as before, that is to say, we join K C and L C B being the angle of discharge we make C L = ψ . C K.

Then the impulse on the second row of buckets is $\frac{1}{g}$ P Q. The total impulse on both rows is therefore

$$\frac{1}{g} . [\text{F G} + \text{P Q}]$$

and the diagram efficiency is

$$e = \frac{2s . (\text{F G} + \text{P Q})}{v^2}$$

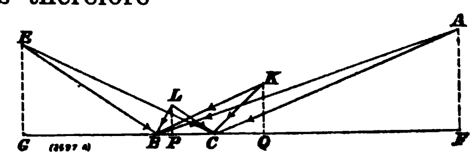


Fig. 47.

If there were a second row of fixed buckets and a third row of moving, there would be a third term between the brackets in the numerator, corresponding to the impulse on the third set of moving buckets.

Considered simply as an exercise in geometry, nothing could be simpler than the determination in this way of the diagram efficiency of a velocity-compounded wheel. Unfortunately, however, the result reached depends upon the values assumed for ψ, the coefficient of velocity, and this is an exceeding variable quantity. If its value be taken from the curve given in Fig. 33, which is based mainly on the results of laboratory experiments, it will be found on trial that even with three rows of moving blades per wheel it is exceedingly difficult to get any appreciable amount of work out of the third row, and yet in practice as many as four, and even five, rows are successfully used.

At the same time, it should be remarked that many designers of velocity-compounded wheels have quite failed to realise higher

efficiences than would correspond to the coefficients taken from the curve in question, Fig. 33, *ante.*

An analysis of successful velocity-compounded wheels shows clearly that the steam is not freely deviated in them, and, in fact, is in certain cases considerably throttled.

With free deviation the edges of the jet must spread as the flow passes through the bucket. The average pressure driving the latter can be calculated, and this must be the same as the effective pressure of the film of steam on its concave face. With high velocity steam and a slowly moving bucket it will be found on calculation that the pressure of the film of steam in question may be quite double the pressure outside the wheel, and hence the edges of the film will tend to spread with a velocity of between 1400 ft. and 1500 ft. per second. Such a velocity represents a considerable amount of kinetic energy, none of which is recoverable as useful work. To prevent this spreading it is essential, therefore, to make the bucket not very much wider than the nozzle. The shrouding, no doubt, does introduce additional friction, but this is more than made good by the restriction it imposes on the lateral spreading of the jet. That this spreading may be considerable is confirmed by Stodola's observation, that with a high velocity jet playing on to a Pelton bucket the stream of steam as it finally issued was as thin as a sheet of notepaper and correspondingly wide.

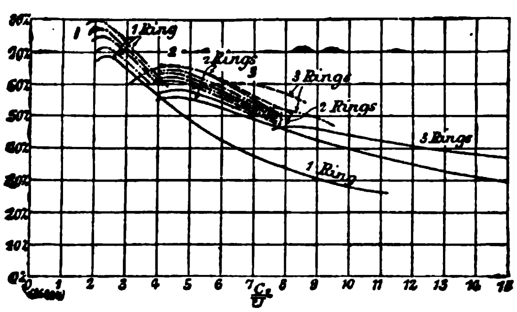


Fig. 48. Efficiency of Velocity-Compounded Wheels.

As the velocity coefficients vary also with the curvature of the bucket, it seems hopeless for the present to fix really satisfactory values for them, and since nothing more than a guess can be made at them it is fully as satisfactory to guess the total efficiency of the whole combination at once, particularly as Dr. Lasche has given a number of curves representing the results obtained in experiments made by the A.E.G., which are reproduced in Fig. 48. The abscissæ represent ratios of steam speed to blade speed.

In Schiffsturbinen higher maximum values than the above for

two, three, and four rows of blades are claimed by Lasche and Bauer to have been realised in practice, viz., 0.72 for two rows of moving blades per wheel, 0.65 for three rows, and 0.52 for four rows of blades per wheel. In the latter case, for maximum efficiency the steam speed should be about eleven times the blade speed.

For electric-light turbines of this type, having two rows of moving buckets per stage and a ratio of blade speed to steam speed of about 0.22, the following proportions are found to give good results:—

—	Angle at Entrance.	Angle at Discharge.	Relative Height at Discharge.
	deg.	deg.	
Nozzle	...	20	1
First row of moving blades ...	28	22	—
First row of fixed blades ...	32	24	- -
Second row of moving blades	39	28	3

A straight line drawn from the extremities of the discharge edge of the last bucket to the corresponding extremities of the nozzle touches the ends of the discharge edges of all the buckets, the height of which can thus be found graphically.

In some cases the bucket height of the last row on discharge is much less than given by the above rule, but the latter represents recent and successful practice.

Some examples of wheels with three and four velocity stages will be found in the description of the turbines of the U.S.S. "Perkins," given in a subsequent Chapter.

CHAPTER X.

REACTION BLADING.

IN the case of reaction blading the diagram of velocities is as represented in Fig. 49. Here $AB = v$ denotes the velocity with which the fluid leaves the fixed guide blades, C B the bucket speed s, whilst $AC = r$ is the velocity of the steam measured relatively to the moving bucket. Just as in the case of an impulse turbine, the bucket turns round the stream of fluid which enters it, and delivers it along the tangent to the bucket at discharge, as indicated by the line C D, which thus represents the direction of the relative velocity of the fluid at discharge.

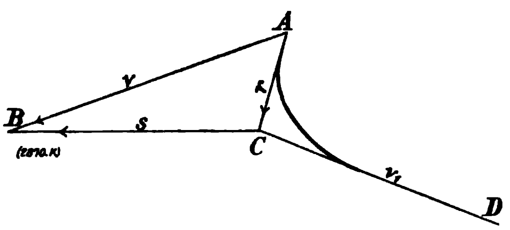


Fig. 49.

In the case of an impulse turbine this relative velocity at discharge is, owing to frictional losses, always less than A C, the relative velocity at entrance, C D being, in fact, often only some 0.7 of A C. In the case of a reaction turbine, on the other hand, C D is always greater than A C. This gain of relative velocity is attained at the expense of a fall of pressure within the moving bucket.

Let the energy liberated (per pound of fluid) by this change in pressure be Q ft.-lb., then the fall in pressure would of itself be capable, theoretically, of generating a velocity equal to $\sqrt{2g \cdot Q}$. On the other hand, if there were no fall of pressure in the bucket and no friction losses, we should have $CD = AC$. Hence the actual kinetic energy with which the fluid finally leaves the moving bucket is derived from two sources—viz., a "carry-in" of kinetic energy equal to $\frac{AC^2}{2g}$, in addition to the work of expansion Q. Hence if there were no frictional losses we should have

$$\frac{CD^2}{2g} = \frac{AC^2}{2g} + Q \qquad . \quad . \qquad . \ (14)$$

In the case of steam turbines, it is often convenient to express quantities of energy in heat units, so that if we put $q = \frac{Q}{778}$, we may also write the above equation in the form

$$\left(\frac{CD}{224}\right)^2 = \left[\frac{AC}{224}\right]^2 + q \quad . \quad . \quad . \quad . \quad . \quad (15)$$

Owing to frictional losses the actual kinetic energy of the escaping fluid is always less than the sum of the two sources generating it, so that in practice it is necessary to write the equation as

$$\left[\frac{CD}{224}\right]^2 = M\left[\frac{AC}{224}\right]^2 + mq \quad . \quad . \quad . \quad . \quad . \quad (16)$$

where M and m are coefficients to be determined by experiment or experience.

By analysing the performance on a trial of a large high-pressure marine steam-turbine, it appears that the value of M is 0.52 and of m 0.90, these figures being determined by the method of least squares. The steam velocities ranged from 180 ft. up to about 300 ft. per second. With these values of the coefficients the results of calculation agree very well, both with the efficiencies observed and with the quantity of steam passed through actual turbines.

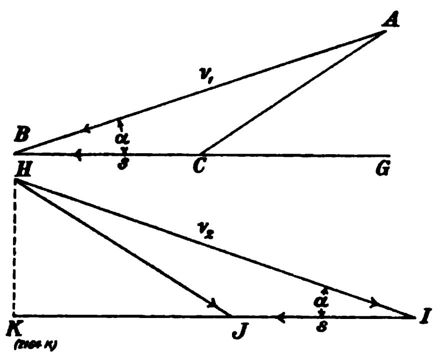


Fig. 50.

The useful work done by a reaction turbine can be determined once C D is known, as follows:—

In Fig. 50 let A B = v_1 be the velocity with which the steam leaves the fixed guide blades. This steam has then a tangential momentum equal to $\frac{w}{g} \cdot v_1 \cos \alpha$, where w denotes the weight passed per second.

The same weight of steam leaves the moving buckets with a relative velocity H I = v_2, and an absolute velocity equal to H J. Its tangential momentum is therefore now equal to $\frac{w}{g}$. K J = $v_2 \cos \alpha - s$. Hence the change of tangential momentum effected per second is

$$\frac{w}{g}(v_1 \cos \alpha + v_2 \cos \alpha - s),$$

and this is numerically equal to the drive on the wheel. The

useful work done per second is equal to this driving force multiplied by the distance moved through in that time, which is of course s, the blade speed.

Hence the useful work done per second is

$$\frac{w}{g} s (v_1 \cos \alpha + v_2 \cos \alpha - s) \text{ ft.-lb.}$$

which may be expressed in heat units by dividing by 778.

In the foregoing it has been assumed that the discharge angle of the moving bucket was the same as that of the fixed blade. In some cases, however, these differ, so that if α_1 be the discharge angle of the fixed blades and α_2 that of the moving blades the useful work done per second becomes

$$\frac{w}{g} s (v_1 \cos \alpha_1 + v_2 \cos \alpha_2 - s) \text{ ft. lb.} \quad . \quad . \quad . \quad (17)$$

The energy expended per pound is equal to the total loss of pressure head between the inlet to the guide blades and the discharge from the moving blades, or, in other words, to the work done on expanding the steam from the pressure p_0 to the pressure p_2.

In general, the group of a reaction turbine consists of more than two rows of blades. A reference to Fig. 50 will make it evident that if another row of fixed blades follows the row of moving ones, there will be a "carry-in" of energy to these fixed blades, represented by $\frac{\text{H J}^2}{2g}$ ft.-lb. The flow through these fixed blades is therefore not solely due to the pressure drop in them any more than it is in the case of the moving blades. It is accordingly irrational to assume that a pair consisting of one row of fixed and one of moving blades can be treated separately and taken as constituting the "stage" or primary element of a reaction turbine. Whether a row of blades is fixed or moving it cannot be disconnected from the preceding row, from which there is always a carry-over of kinetic energy, which assists in generating the actual velocity of discharge. The rational method of analysing a reaction turbine is therefore to divide it up into its groups, each group consisting of rows of blades all of the same height, and each group is, in its turn, rationally divisible into stages, each stage consisting of one row of blades, whether that row be moving or fixed.

It is the fact just mentioned which so greatly complicates the theory of the reaction steam turbine, and the discussion of the

flow of steam through a group of blades is therefore reserved for another Chapter, in which the equations of flow will be established and numerical results arrived at.

The investigation is in itself somewhat lengthy, but fortunately the results can be reduced to some relatively simple formulas and curves, the use of which is explained in the Chapters on the Proportioning of a Reaction Turbine.

Radial-Flow Turbines.

The elementary theory of the reaction turbine as given above has relation only to axial-flow machines. Where, as in some steam turbines, the flow through the blades is radial in place of axial, the theory requires modification

A radial-flow turbine may be considered as more or less analogous to an inverted centrifugal pump, and thus the flow of the fluid is partially due to centrifugal forces. If a radial-flow turbine be of the outward-flow type, these centrifugal forces promote the passage of the fluid, whilst with the inward-flow type they impede it.

With a radial-flow turbine the work done by a pound of fluid in passing through a row of moving blades is equal to the angular velocity of the wheel multiplied by the change effected in the moment of momentum of the fluid.

Thus, let v_1 be the absolute tangential velocity of the pound of fluid just before it enters the row of moving blades, its momentum is then $\frac{v_1}{g}$. Let its distance from the centre of rotation be r_1, then $\frac{r_1 v_1}{g}$ is called its moment of momentum.

At the instant at which it leaves the row of moving blades let its absolute velocity be v_2 and its distance from the centre of rotation be r_2. Its moment of momentum is now

$$\frac{v_2 r_2}{g}.$$

This quantity will be negative if v_2 is oppositely directed to v_1.

Taking due account of the sign of v_2, the change in the moment of momentum will thus be

$$\frac{v_1 r_1}{g} - \frac{v_2 r_2}{g}.$$

By an elementary principle in mechanics, a change in the moment of momentum of a body round a given point can only be brought about by a torque numerically equal acting round the same point.

Hence the torque T which has acted on the pound of fluid is

$$T = \frac{v_1 r_1}{g} - \frac{v_2 r_2}{g},$$

and this, since action and reaction are equal and opposite, is also the torque on the wheel. The useful work done by the latter is equal to the torque on it, multiplied by the angular velocity ω. Whence the useful work done per pound of fluid passed is

$$T \varpi = T . \frac{2 \pi \text{R.P.M.}}{60} = \frac{2 \pi . \text{R.P.M.}}{60 . g} . [r_1 v_1 - r_2 v_2],$$

where, as already stated, $\frac{r_2 v_2}{g}$ may be, and indeed usually is, a negative quantity. In that case it is generally more convenient to neglect the sign of v_2 and write $T = \frac{r_1 v_1}{g} + \frac{r_2 v_2}{g}$.

The further development of the theory of the radial-flow turbine is left to another Chapter.

CHAPTER XI.

REACTION TURBINE DESIGN.

IN a previous Chapter we have shown how it is possible to arrive rapidly at the general proportions of a reaction turbine to give a given output and given steam consumption per horse-power hour. The latter is, however, dependent upon a number of variable factors, such as the leakage through the dummies and glands, and on the amount of clearance allowed over the blade tips. The loss from the latter source may be very considerable where the clearance is a substantial fraction of the blade height, as it often is at the high-pressure end of a marine turbine, so that the actual efficiency may be materially less in such cases than would be estimated from the curve in Fig. 43, *ante*. Speaking roughly, an increase of 1 per cent. in the tip clearances increases the consumption per horse-power hour by 2 per cent.

Some valuable experiments on the effects of tip clearances have been carried out at the works of the Brush Electrical Engineering Company, Loughborough, by Messrs. W. Chilton and J. M. Newton, to whom the author is indebted for the results plotted in Fig. 51. Here the radial clearance over some ⅜ in. standard Parsons blades was increased from zero up to ⅛ in., and the steam passed through collected and weighed. The curve shows that the leakage plots against the clearance as a straight line, and it follows accordingly that the leakage flow cuts straight across the tops of the blades and is not sensibly deflected by the main flow.

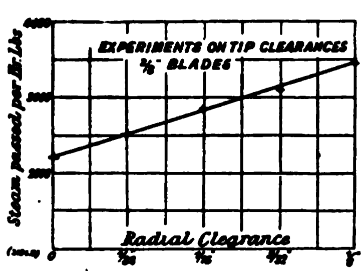


Fig. 51.

As stated in Chapter III., the mean angle of discharge of standard Parsons blades gauged to one-third pitch is about 18° 20′,

so that if Ω^1 denotes the area available for flow when there is no tip leakage we have, if h be the blade height,

$$\Omega^1 = \frac{h\,C \sin(18^\circ\ 20')}{144} \text{ sq. ft.}$$

Where C denotes the mean circumference in inches. If Ω denote the effective area taking leakage into account

$$\Omega = \frac{(h + \sigma c)\,C \sin 18^\circ\ 20'}{144} \text{ sq. ft.} \quad . \quad . \quad . \quad (21)$$

where $\sigma = \dfrac{1}{\sin a} - 1$, a being the discharge angle of the blades. For normal blades $\sigma = 2.15$, for semi-wing blades 1.1, and for wing blades about 0.6. Here c denotes the tip clearance.

Of the total weight w, which passes through a group of blades, the proportion which does useful work, is

$$\frac{h - c}{h + \sigma c} \cdot w.$$

In the previous Chapter an approximate expression was given for the velocity with which the steam enters the first row of blades of a reaction turbine. The expression is practically an empirical one, but gives a very fair approximation to the truth in the case of a turbine designed on standard lines. Actually, however, this velocity is not constant, but varies with the ratio of blade speed to steam speed, though it is nearly independent of the pressure below the governor valve and of the vacuum in the condenser. In short, experiment shows that the steam speed into the first row of blades of a reaction turbine is nearly constant at all loads provided the blade speed is kept constant.

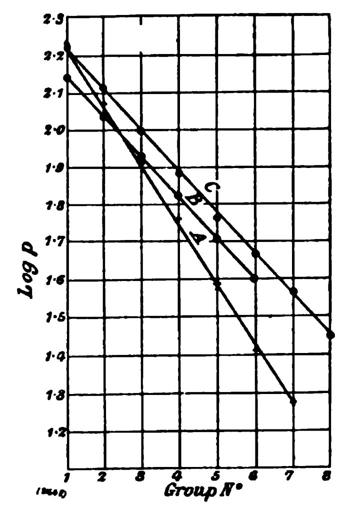


Fig. 52.

It is further found that if the blade height increases from group to group in a given geometric ratio, the pressure before each successive group diminishes in nearly the same ratio, at least at the high-pressure end of the turbine. If this law were strictly true, the logarithms of the pressures plotted against the group numbers should lie on a straight line, and, as will be seen from Fig. 52, this is very nearly true. Here along the line A have

been plotted the pressures observed along the high-pressure turbine of a large warship, in which the blade heights increased in the ratio of $\sqrt{2}$ to 1.

Along line B have also been plotted the logarithms of the pressures in front of each group of the high-pressure turbine of a large Atlantic liner. In this case the ratio between successive blade heights was $\sqrt[3]{2}$, and again it will be seen from the slope of the curve that this is approximately the ratio between successive pressures, though it will be seen that this pressure ratio is rather better represented by 1.29 than by $\sqrt[3]{2}$. The pressures, of which the logarithms have been plotted along line C, were observed on the "Mauretania," and here also successive points fall very accurately on a straight line, so that, as before, the pressures vary in a geometric ratio, which is nearly the inverse of that of successive blade heights.

In the case of low-pressure turbines less uniform results are obtained, partly owing to the "sticktion," characteristic of ordinary vacuum gauges, but mainly to the fact that an increase or decrease of vacuum affects in general the last groups of blades only, and is without influence on the distribution of pressures higher up the turbine.

In the Chapters on Flow Through Groups of Blades it is proved that a rational expression can be found for the velocity of inflow into the group of a turbine when the ratio of pressures at the beginning and end of the group are known, in addition to p_0 the pressure and V_0 the specific volume of the steam just before it enters the group.

From what has been stated above it will be seen that with a blade-height ratio of $\sqrt{2}$ to 1 the ratio of the pressures $\frac{p_0}{p_N}$ will be about 1.41, whilst when the blade heights increase in the ratio, $\sqrt[3]{2}$ to 1, the ratio of $\frac{p_0}{p_N}$ is also nearly $\sqrt[3]{2}$ to 1, but is perhaps better taken as 1.29. On this basis the following table has been drawn up showing v_0, the velocity of inflow into the first row of an ordinary reaction turbine for different values of $\delta = \frac{\text{blade speed}}{v_0}$.

This table is based upon the assumption that $p_0 V_0$ is about 455, which is very approximately the case in practice with high-pressure turbines.

TABLE V.

N = Total Number of Rows in Group (both Fixed and Moving).	$v_0 = \frac{w V_0}{\Omega}$ = velocity of steam at entrance to group.					
	$x = \frac{p_0}{p_N} = \frac{\text{Pressure at Entrance}}{\text{Pressure at Discharge}} = 1.29.$			$x = \frac{p_0}{p_N} = \frac{\text{Pressure at Entrance}}{\text{Pressure at Discharge}} = \sqrt{2} = 1.41.$		
	$\delta = \frac{\text{Blade Speed}}{v_0} = 0.4$	$\delta = \frac{\text{Blade Speed}}{v_0} = 0.5$	$\delta = \frac{\text{Blade Speed}}{v_0} = 0.6$	$\delta = \frac{\text{Blade Speed}}{v_0} = 0.4$	$\delta = \frac{\text{Blade Speed}}{v_0} = 0.5$	$\delta = \frac{\text{Blade Speed}}{v_0} = 0.6$
	ft. per sec.	ft. per sec.	ft. per sec.	ft. per sec.	ft. per sec.	ft. per sec.
4	482.1	469.7	459.7	528.6	517.1	508.2
6	398.8	388.2	379.4	440.0	430.7	422.3
8	347.7	338.3	330.5	385.5	376.9	369.0
10	312.3	303.7	296.6	347.2	339.2	338.1
12	285.9	277.9	271.4	318.5	310.9	303.6
14	265.2	257.7	251.6	295.9	288.7	282.3
16	248.4	241.5	235.7	277.5	270.7	264.6
18	234.5	227.9	222.4	262.1	255.7	249.8
20	222 6	216.4	211.2	249.1	242.9	237.3
22	212.5	206.4	201.4	237.8	231.9	226.5
24	208.5	197.8	193.0	227.9	222.2	217.0
26	195.6	190.1	185.3	219.2	213.7	208.7
28	188.3	183.2	178.8	211.4	206.1	201.2
30	182.3	177.1	172.8	204.4	199.2	194.5
32	176.6	171.6	167.4	198.0	193.0	188.4
34	171.4	166.5	162.4	192.2	187.3	182.8
36	166.6	161.8	157.9	186.9	182.1	177.8

If we take the marine turbine, of which the general proportions were got out on page 63, we found $v_0 = 222.5$ approximately whilst s the blade speed $= \frac{46\ \pi \times 465}{60 \times 12} = 92.5$ ft. per second, so that

$$\delta = \frac{92.5}{222.5} = 0.415 \text{ nearly.}$$

As there are 24 rows in the first group, the corrected steam speed will, it appears from Table V., be about 227 ft. per second.

In the same Chapter the area available for flow was taken as one-third the annulus between the casing and drum. The true effective area, as given on page 84, is

$$(h + \sigma c)\frac{\mathrm{C} \cdot \sin 18^\circ 20'}{144}.$$

In marine practice the clearance allowed is often 8 mils per foot of drum diameter + 5 mils per inch of blade height, so that as h is 1 in., about 35 mils will be the value of c for the first row of blades, and thus we get

$$\Omega = \frac{1.035 \times 144.5 \times 0.3145}{144} = 0.326 \text{ sq. ft.}$$

The volume passed per second is $v_0 \times 0.326 = 227 \times 0.326 = 74$ cub. ft., which, at 2.61 cub. ft. per lb., gives a flow of 28.6 lb. per second, which is a little less than the approximate estimate previously made on page 63.

In Chapter XIV., on Groups of Blades with Constant Discharge Angle, it is proved that the work W done by a group of blades per pound passed per second is given by the relation

$$\mathrm{W} = \frac{h - c}{h + \sigma c} \cdot \frac{s}{32.2} \cdot \left\{v_0 (\mathrm{N} + 1.05 \log x)\, b \cos \alpha - \frac{\mathrm{N}s}{2}\right\},$$

where s denotes the blade speed, N the number of rows of blades in the group, and x is the ratio $\frac{p_0}{p_n}$ of the pressure of the steam at entrance into, to the pressure of the steam at discharge from, the group, whilst α denotes the angle of discharge, and b a coefficient, which is 1.093 for $x = 1.41$, and 1.065 for $x = 1.29$.

Taking standard blades set to one-third pitch, the formula given above simplifies to

$$\mathrm{W} = \frac{h - c}{h + \sigma c}\ \frac{s}{32.2} \cdot \left\{1.093\, v_0 (\mathrm{N} + 0.16) - \frac{\mathrm{N}s}{2}\right\} \text{ for } x = 1.41$$

and

$$\mathrm{W} = \frac{h - c}{h + \sigma c}\ \frac{s}{32.2} \cdot \left[1.056\, v_0 (\mathrm{N} + 0.13) - \frac{\mathrm{N}s}{2}\right] \text{ for } x = 1.29.$$

Hence the work done by the first group of blades in the present case will be

$$\frac{1 - 0.035}{1 + 0.075} \times \frac{92.5}{32.2} \left\{1.093 \times 277 \times 24.16 - \frac{24 \times 92.5}{2}\right\}$$

= 12,595 ft.-lb. per second per lb. of steam passed. Since the

weight passed is 28.6 lb. per second, the horse-power developed by the first group will be

$$28.6 \times \frac{12{,}595}{550} = 655 \text{ indicated horse-power.}$$

To obtain the same average of power from each of the remaining groups, experience shows that the condenser pressure should be equal to about $\frac{p_0}{106} = 1.65$ lb. absolute; equivalent to a vacuum of about 26.63 in., so that the horse-power developed will be $12 \times 655 = 7870$. An increase of the vacuum to 28 in. will, as the correction curve, Fig. 36, page 47, shows, increase this figure to 8420 indicated horse-power, which will be fully equivalent to 8000 shaft horse-power.

The total steam consumption includes, however, not only the steam through the blading but also that which leaks through the dummies.

These dummy losses can be calculated by the formula established in Chapter XVIII., viz. :—

$$\text{Weight of steam through dummy per second} = 68\,\Omega \sqrt{\frac{p_0\left(1 - \frac{1}{x^2}\right)}{V_0\,(N + \log_\epsilon x)}}$$

where Ω denotes the area (measured in square feet) available for the flow of steam, p_0 the absolute pressure in pounds per square inch of the steam on the high-pressure side of the dummy, and V_0 the corresponding specific volume of the steam ; N denotes the number of points at which the steam is wire-drawn, or, in other words, the number of rings in the dummy, whilst x denotes the ratio of the pressure, at entrance to the dummy, to the pressure on discharge. When the three-shaft arrangement of turbines is adopted, x for the high-pressure dummy may be taken as very approximately 4, so that the formula for flow becomes

$$\text{Weight discharged} = 68\,\Omega \sqrt{\frac{p_0\left(\frac{15}{16}\right)}{V_0\,(N + 1.38)}}.$$

If we take N as 30, the weight discharged from the high-pressure dummy is then equal to $11.76\,\Omega \sqrt{\frac{p_0}{V_0}}$.

To find Ω it is necessary to know the dummy diameter which

depends upon the relative thrusts of the screw and of the steam. The axial thrust of the steam on the rotor-blades is given very approximately by the formula

Thrust of steam $= \frac{p_0 \, A \, G}{2} \cdot \left(1 - \frac{1}{j}\right)$ where p_0 is the pressure in front of the first row of blades of the turbine, A the area in square inches of the annulus between the drum and the casing at the first row of blades, and G the number of groups in the turbine, whilst j is the ratio of successive blade heights. In the present case we have $p_0 = 175$, A = 144.5 sq. in., G = 4, and $j = \sqrt{2}$. Whence we get

$$\text{Thrust of steam} = 14{,}819 \text{ lb.}$$

This is more than balanced by the thrust of the propeller. The high-pressure turbine in fact develops 2667 shaft horse-power, so that, assuming the propulsive efficiency to be 0.50, and the designed speed 21 knots, or 35.5 ft. per second, we have for the propeller thrust T the equation

$$T \times 35.5 = 0.50 \times 2667 \times 550 = 1334 \times 550$$

or

$$T = 20{,}667 \text{ lb.}$$

The difference between this and the steam thrust is 20,667 − 14,819 = 5848 lb., which must be compensated for by making the dummy diameter less than that of the turbine main drum. The pressure being 175 lb. per sq. in. at the entrance to the dummy and about 44 lb. absolute on the discharge side, the effective pressure is 131 lb. and the area required is $\frac{5848}{131} = 44.6$ sq. in.

The drum diameter being 45 in., it has an area of 1590.4 sq. in., and the dummy area must be 44.6 in. less, or 1545.8 sq. in. Hence the dummy diameter should be $44\frac{3}{8}$ in., the corresponding circumference being 139.4 in.

Dummy clearances are commonly about 5 mils per foot of diameter—say, 20 mils in this case. Hence the area available for flow through the dummy is $\frac{139.4}{144} \times 0.02$ sq. ft. = 0.0194 sq. ft.

All the data necessary for calculating the dummy leakage are now known, and we get the weight discharged per second

$$= 68 \times 0.0194 \sqrt{\frac{175}{2.602} \cdot \frac{15}{16 \times 31.38}} = 1.87 \text{ lb. per second.}$$

This is the loss through the high-pressure dummy, but as this leakage passes into the low-pressure turbines at a point where the pressure is about 20 lb. above the atmosphere, only one-third of it, or 0.62 lb., is net loss. This brings up the steam consumption per second to 29.22 lb., and to this must now be added the loss from the low-pressure dummies.

The propeller thrust will be the same as before, and the same formula may be used for the steam thrust, though, owing to the presence of the wing blades and the rapid fall of pressure through them, it is less accurate here than as applied to the high-pressure turbine.

The pressure at entrance to the low-pressure turbine, allowing for losses in the connecting-pipe, may be assumed as equal to 42 lb. absolute, whilst the area A of the annulus will be 204.2 sq. in., the drum diameter being 64 in., and the blade height 1 in. Hence the formula

$$\text{Steam thrust} = \frac{p_0 \text{ A. G.}}{2}\left(1 - \frac{1}{j}\right)$$

becomes

$$\text{Steam thrust} = \frac{42 \times 204.2 \times 8}{2} \times 0.293 = 10{,}052 \text{ lb.}$$

Hence the thrust to be taken by the dummy is 20,667 − 10,052 = 10,615 lb. Taking the effective pressure as 41 lb. per sq. in., the area needed is 259 sq. in. nearly. The drum cross-section has an area of 3217 sq. in., hence the cross-section of the dummy must have an area equal to 3217.0 − 259 = 2958 sq. in., equivalent to a dummy diameter of $61\frac{3}{8}$ in., the corresponding circumference being 192.8 in. Taking, as before, the clearance as equal to 5 mils per foot of diameter, or 25 mils in all, the area available for flow through the dummy is $\dfrac{192.8 \times 0.025}{144} = 0.0334$ sq. ft.

Taking p_0 as 42, V_0 as 9.6 cub. ft. per pound, and x as 34, approximately, we get for the flow through one dummy

$$\text{Weight discharged per second} = 68 \times 0.0334\sqrt{\frac{42}{9.6\,(19.05)}}$$

= 1.086 lb. per second through one dummy, or 2.172 lb. through two. As the low-pressure turbines do two-thirds of the work, the net loss is $2.172 \times \frac{2}{3} = 1.45$ lb. per second, which brings the steam consumption up to 29.22 + 1.45 = 30.67 lb. per second.

The gland leakage may be calculated in the same way. A standard marine gland is illustrated in Chapter XVIII ; the clearance allowed is about 8 mils per foot of shaft diameter. Taking the shaft as 12 in. in diameter at the glands, and applying the formula already given, the total leakage at the two high-pressure glands works out at 0.188 lb. per second, about half of which can be saved if the "leak off" is led into the low-pressure turbine half-way down the casing. In the case of the low-pressure turbines the loss from each gland amounts to 0.0147 lb. per second, or 0.06 lb. for the four. Thus the aggregate loss from the glands may be taken as 0.15 lb. per second, making the total steam consumption 30.82 lb. per second, so that the consumption per shaft horse-power is $\frac{30.82 \times 3600}{8000} =$ 13.85 lb. per hour.

The consumption estimated from Fig. 43, page 50, was about 13 lb. per shaft horse-power; the difference is mainly due to the relatively large clearance over blade tips.

CHAPTER XII.

REACTION TURBINE DESIGN—(*continued*).

THE plan commonly adopted of making the blade heights from group to group of a reaction turbine increase in geometric ratio does not lead to an equal partition of the available energy between the groups. Considerations of symmetry indicate that an equipartition of energy between the different groups should be favourable to economy, though there is no reason for supposing that any great advantage can thus be realised. Nevertheless it is of some interest to work out what are the blade heights which would lead to an equal division of energy between the different groups. This can be done pretty simply if the blades of a group are "gauged" so as to keep the steam velocity the same in each row.

Let it be assumed that the steam below the governor valve is at 165 lb. absolute and dry, and that the vacuum is 28 in., and that λ is 106,000, as in the case of the 2000-kw. reaction turbine, of which the principal dimensions were obtained in Chapter VII. With this value of λ the efficiency ratio reckoned at the turbine shaft is, from Fig. 43, about $67\frac{1}{2}$ per cent., which will correspond, say, to an indicated efficiency ratio of about 0.705. Hence, instead of turning into indicated work 321 B.Th.U., as a perfect turbine would do, the actual turbine subtracts from the steam only about $321 \times 0.705 = 226$ B.Th.U. per lb. Its total heat originally was about 1193 B.Th.U., so that on exhaust at a 28-in. vacuum it will contain $1193 - 226 = 967$ B.Th.U. per lb. From the Mollier diagram it appears therefore that the steam on final discharge is about 86 per cent. dry, so that its specific volume $V_x = 340 \times 0.86 = 292.4$ cub. ft. per lb., the pressure p_x being 0.98 lb. absolute. Initially the pressure p_0 was 165, whilst $V_0 = 2.757$.

The expansion takes place very approximately according to the law

$$p V^\gamma = p_0 V_0^\gamma,$$

where

$$\gamma = \frac{\log p_0 - \log p_x}{\log V_x - \log V_0}.$$

Thus

$$\gamma = \frac{2.21748 - \bar{1}.99123}{2.46598 - 0.44044} = \frac{2.22625}{2.02554} = 1.0991.$$

We may further write

$$A_x = \frac{V_x}{V_0} \cdot \frac{p_x}{p_0} = \frac{\rho_x}{x_x}$$

Substituting for V_x, V_0, and p_x, p_0, we get

$$A_x = 0.62993.$$

If there are to be twelve groups, with the energy equally divided between them, we have, by the rule given in Chapter VIII., page 69.

$$12\ A_1 - 11 = A_x, \text{ where } A_1 = \frac{p_1\ V_1}{p_0\ V_0} = \frac{\rho_1}{x}$$

So that

$$A_1 = \frac{11.62993}{12} = 0.96916$$

and then

$$A = \frac{p_2\ V_2}{p_0\ V_0} = 2\ A_1 - 1$$

$$A_3 = \frac{p_3\ V_3}{p_0\ V_0} = 3\ A_1 - 2, \text{ \&c., \&c.,}$$

and we thus get the values of A given in the first column of Table VI., page 94.

Having these values of A, we can find the value of V at any group by the relation

$$V = V_0\left(\frac{1}{A}\right)^{\frac{1}{\gamma - 1}} = V_0\left(\frac{1}{A}\right)^{\frac{1}{0.0991}}$$

whence

$$\log V = \log V_0 + \frac{1}{0.0991} \cdot \log \frac{1}{A}.$$

The calculation is given in detail in the table. If the corresponding pressures are required for any purpose, we have

$$\log p = \log p_0 - \frac{1.0991}{0.0991} \cdot \log \frac{1}{A} \quad . \quad . \quad . \quad . \quad . \quad (22)$$

Table VI., page 94, gives the specific volume in the front of each group if the energy is equally divided between the twelve groups, and these volumes will be the same whatever the output of the turbine.

TABLE VI

Group Number.	A	Log A.	$\text{Log}\,\frac{1}{A}$.	$\frac{1}{0.0991}\log\frac{1}{A}$.	Log V.	V cub. ft. per lb.
1	1.0000	0.00000	0.00000	0.0000	0.4404	2.757
2	0.9692	$\bar{1}.98641$	0.01359	0.1368	0.5772	3.780
3	0.9383	$\bar{1}.97234$	0.02766	0.2791	0.7191	5.237
4	0.9075	$\bar{1}.95785$	0.04215	0.4253	0.8657	7.340
5	0.8766	$\bar{1}.94280$	0.05720	0.5772	1.0176	10.41
6	0.8458	$\bar{1}.92727$	0.07273	0.7339	1.1743	14.94
7	0.8150	$\bar{1}.91116$	0.08884	0.8965	1.3369	21.72
8	0.7841	$\bar{1}.89437$	0.10563	1.0658	1.5062	32.08
9	0.7533	$\bar{1}.87697$	0.12303	1.2413	1.6817	48.05
10	0.7225	$\bar{1}.85884$	0.14116	1.4236	1.8640	73.26
11	0.6916	$\bar{1}.83985$	0.16015	1.6159	2.0563	113.9
12	0.6608	$\bar{1}.82007$	0.17993	1.8155	2.2559	180.3
—	0.6299	$\bar{1}.79929$	0.20071	2.0255	2.4659	292.4

If, as before in Chapter VII., the turbine is to develop 2000 kw. at 1500 revolutions, and we take the mean diameter of the blade path as 19 in., the blade height required will be $\frac{3}{4}$ in. as before, and the remaining blade heights, neglecting the effect of tip leakage, can be determined from the volumes given in the last line of the table.

Thus, assume provisionally that the mean blade path of all the groups is 19 in., in diameter, so that the blade heights at successive groups will be directly proportional to these volumes. Of course this assumption is really an impossible one, as the blades soon become too long. Nevertheless the theoretical blade heights thus calculated can be made the basis from which the height of blade required on the necessary enlargements of the drum diameter can be calculated. Taking these heights as proportional to the specific volume we thus get the table on page 95.

The great range of the blade heights, as tabulated on the next page, points to the advisability of constructing the drum in four diameters.

This is indeed practically unavoidable where a high speed of

rotation is to be used, since a drum of sufficient diameter to give the steam way theoretically required at the low-pressure end would be subject to very high centrifugal stresses. With a four-diameter rotor, however, the low-pressure section can be constructed as a solid disc, and is thus able to withstand safely a very high speed.

Group Number.	Provisional Blade Height in Inches.	Group Number.	Provisional Blade Height in Inches.
1	0.75	7	5.92
2	1.03	8	8 74
3	1.425	9	13.09
4	2.00	10	19 92
5	2.84	11	31.0
6	4.06	12	49.4

Each of the groups in the provisional table above comprises twelve rows of moving blades. If then the low-pressure groups as finally settled are given two moving rows each, the mean diameter of the group must be increased in the ratio of 1 to $\sqrt{\frac{12}{2}}$, so that the mean diameter of Group 12 will be $19 \times \sqrt{6} = 46.6$ in. The corresponding blade height is then obtained by dividing the provisional value by 6, so that the calculated height for Group 12 becomes 8.21 in.

If Group 11 be also placed on this disc, its blade height as a first approximation will be $\frac{31.0}{6} = 5.17$ in.

With the same disc diameter as Group 12, however, the mean diameter of Group 11 will be less, so that the above figures should be increased in the ratio of $46.6 : (46.6 - 8.21 + 5.17)$: or blade height for Group 11

$$= \frac{5.17 \times 46.6}{43.56} = 5.50 \text{ in.}$$

A further adjustment might now be made to keep the ratio of blade speed to steam speed the same as for Group 12, but small variations in this from group to group are of little importance.

Coming next to the second intermediate section, if it is decided to give each group here four rows of moving blades the mean diameter should be $19 \times \sqrt{\frac{12}{4}} = 32.9$ in. The blade height required

for Group 10 will then be one-third of that given in the provisional table, so that this group will have four rows of moving blades 6.64 in. high. The blade heights for the other groups, after adjustment, for their smaller mean diameter in the fashion explained above are as follow :—

Group number	7	8	9	10
Calculated blade height, inches ...	2.29	3.28	4.57	6.64

Similarly, allowing eight rows of moving blades per group on the first intermediate drum, we get $19 \times \sqrt{\frac{12}{8}} = 23.3$ in. as the mean diameter of Group 6, and the corresponding blade heights after adjustment are as follow :—

Group number	4	5	6
Calculated blade height, inches ...	1.40	1.96	2.71

For the high-pressure section of the drum the blade height of Group 2 after adjustment for its mean diameter being greater than 19 in. becomes 1.015 in. and that of Group 3 ; 1.376 in.

The figures thus calculated may, if desired, be further adjusted to allow for the fact that tip clearances are relatively more important the shorter the blades, and account may also be taken in the same way of the fact that owing to dummy leakages the weight of steam passed through the blading increases with each successive section of the turbine. Since, however, blade heights have to be made even dimensions in the end, refinements of calculation should not be pushed too far.

The method above given for dividing up the available energy equally between the different groups suffices if the blading is gauged so that the steam velocity remains constant throughout each individual group, but if, as is frequently done, the discharge angle of the blades is kept constant, blade heights proportioned as above will not then give an equal partition of the available heat. With a constant blade angle the area available for flow remains the same from one end of a group to the other, and as the steam expands on its passage it follows that its velocity at the last row is substantially higher than it was at the first row.

In these conditions the velocity of inflow into a group depends not only on the available heat but also on the density of the steam, the fact being that low-pressure steam, in developing a given amount of energy, increases its volume proportionately more than high-pressure steam does. With the same velocity at the first row of a group the low-pressure steam will have a higher velocity at discharge, and hence a larger expenditure of energy will be necessary to maintain its flow.

By the use of the curves, Figs. 53 to 56, on the next page, of which the theory is given later, it is possible to proportion blade heights so that an equipartition of energy is secured also in this case. The method of calculation is set forth in Table VII., on page 99.

In the first row of this table the values of V, taken from Table VI., are set down, and under them the corresponding values of the steam pressure p, as determined by equation (22), page 93. From these latter figures we get the corresponding value of $x = \frac{\text{pressure in front of group}}{\text{pressure behind group}}$, and with this value of x we can, on assuming δ_N, which denotes the ratio $\frac{\text{blade speed}}{\text{steam speed at discharge}}$, find from Fig. 53 values of ϕ and μ. In this case δ_N has been taken as 0.4. If next the total number of rows of blades per group is assumed, the velocity v, with which the steam is finally discharged from the group, can be calculated by the relation

$$\left(\frac{v}{100}\right)^2 = \frac{\phi \times p\,\mathrm{V}}{\mathrm{N} + \mu}$$

Having v; the necessary value of s to give $s = 0.4\,v$ can be found, and if we take the speed as 1500 R.P.M. the value of d, the mean diameter of the blading in inches is equal to $\frac{s \times 12 \times 60}{1500 \times \pi} = \frac{s}{6.54}$. These values of d are tabulated along line 12, and they would, it will be seen, involve a drum having numerous steps.

The approximate blade heights, in inches (to be adjusted afterwards), are given by

$$h = 0.6 \times \mathrm{R.P.M.} \times \frac{w\,\mathrm{V}}{v\,s}$$

where w denotes the weight passed per second. Taking w as

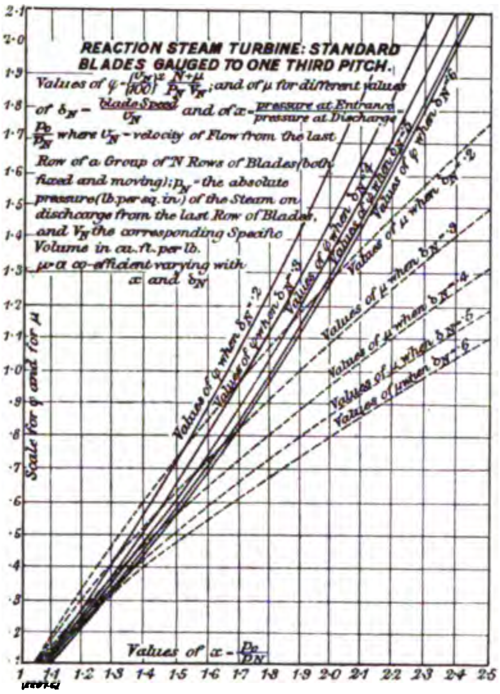


Fig. 53.

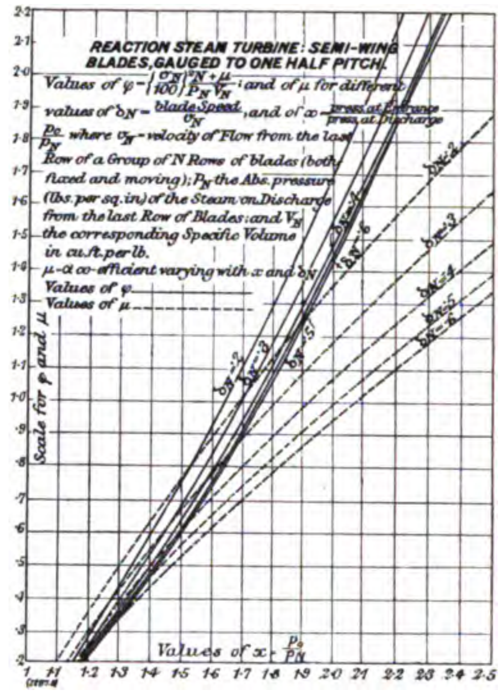


Fig. 54.

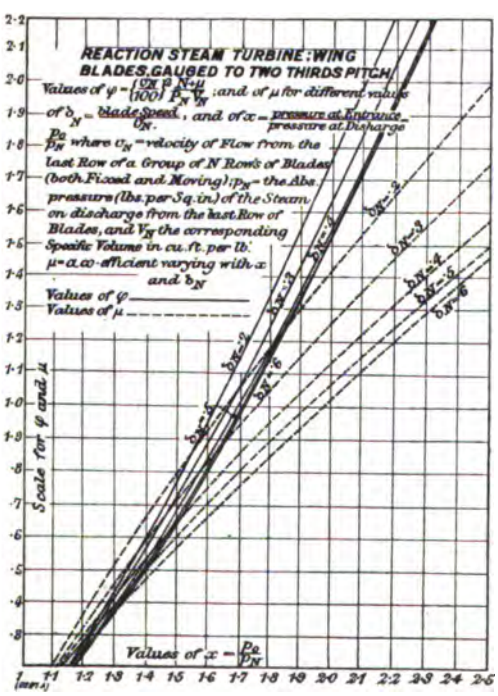


Fig. 55.

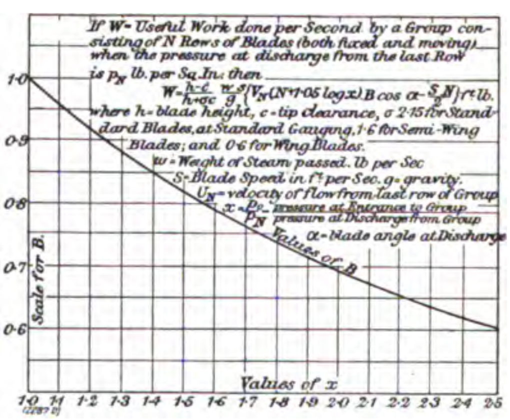


Fig. 56.

Figs 53 to 56. Curves for Analysing Groups of Reaction Blading with Constant Discharge Angles.

TABLE VII.—THEORETICAL PROPORTIONS OF TURBINE (WITH CONSTANT BLADE ANGLES) TAKING STEAM AT 165 LB. ABSOLUTE, AND EXHAUSTING AT A 28-IN. VACUUM. WEIGHT OF STEAM PASSED PER SECOND = 7.3 LB. REVOLUTIONS PER MINUTE = 1500.

—	Group Number	1	2	3	4	5	6	7	8	9	10	11	12
1	Specific volume on discharge V, cub. ft.	3.780	5.237	7.340	10.41	14.94	21.72	32.08	48.05	73.26	113.9	180.3	292.4
2	Pressure on discharge p, lb. per sq. in.	116.7	81.41	56.23	38.29	25.75	17.06	11.12	7.132	4.486	2.763	1.667	0.9800
3	$x = \frac{p_0}{p_N}$	1.415	1.432	1.449	1.471	1.486	1.509	1.535	1.560	1.589	1.622	1.658	1.701
4	ϕ (from Fig. 53)	0.485	0.509	0.530	0.561	0.580	0.610	0.652	0.684	0.724	0.773	0.826	0.888
5	μ (from Fig. 53)	0.492	0.509	0.523	0.550	0.560	0.580	0.602	0.625	0.650	0.680	0.711	0.749
6	Total number of rows . N	24	24	24	16	16	16	8	8	8	4	4	4
7	p V (line 1 × line 2)	441.1	426.4	412.7	398.6	384.7	370.6	356.7	342.7	328.6	314.7	300.5	286.5
8	$\frac{v^2}{100} = \frac{\phi \times p\,V}{N \times \mu}$	8.737	8.855	8.921	13.51	13.45	13.63	27.04	27.18	27.50	52.00	52.69	53.57
9	Velocity at discharge v, ft. per second	295.6	297.6	298.6	367.5	367.2	369.2	520.0	521.3	524.4	721.0	725.9	731.9
10	Blade speed $s = 0.4\,v$	118.2	119.0	119.5	147.0	146.9	147.7	208.0	208.5	209.7	288.8	290.3	292.7
11	Approximate blade height, in.	0.71	0.97	1.35	1.27	1.820	2.63	1.95	2.80	4.40	3.60	5.60	9.00
12	Calculated mean diameter, in.	18.1	18.2	18.3	22.5	22.5	22.6	31.8	31.9	32.1	44.1	44.4	44.7
13	Tip clearance, mils	20	25	30	30	30	35	40	45	55	65	75	90
14	B (from Fig. 56)	0.844	0.839	0.834	0.827	0.822	0.816	0.807	0.800	0.792	0.784	0.773	0.762
15	(N + 1.05 log x)	24.185	24.164	24.168	16.176	16.181	16.187	8.195	8.206	8.211	4.221	4.228	4.243
16	v . B . (N + 1.05 log x) cos α	5719.2	5726.6	5713.7	4666.0	4635.3	4626.7	3263.9	3248.1	3236.6	2262.0	2251.9	2245.7
17	$\frac{N\,s}{2}$	1418.4	1428.0	1434.0	1176.0	1175.2	1181.6	832.0	834.0	838.8	577.6	580.6	585.4
18	Q = (line 16 − line 17)	4300.8	4298.6	4279.7	3490.0	3460.1	3445.1	2431.9	2414.1	2397.8	1684.4	1671.3	1660.3
19	Work done per pound of steam $= W = \frac{h - c}{h + \sigma c} \cdot \frac{s\,.\,Q}{25{,}050}$. B.Th.U.	18.68	18.85	19.04	19.02	19.27	19.56	18.94	19.11	19.30	18.35	18.57	18.80

7.3 lb. per second, the approximate value of h is as tabulated along line 11.

The tip clearance c may be taken as 1 mil for each inch of drum diameter plus 5 mils for each inch of blade height. The values of c are tabulated along line 13. Then, the fraction of the steam which does useful work in a group is equal to $\frac{h - c}{h + \sigma c}$ where σ = 2.15 for normal blades.

Hence, if we take from Fig. 56 the values of B corresponding to the values of x in line 3, we can calculate the work done per pound of steam passed by each group, as shown in lines 19, by the relation proved in Chapter XIV., viz.—

Useful work done per pound of steam passed is given by

$$W = \frac{h-c}{h+\sigma c} \cdot \frac{s}{778 \, . \, g} \cdot \left\{ v \; (N + 1.05 \log x) \; B \cos \alpha - \frac{s \, \mu}{2} \right\} \text{ B.Th.U.,}$$

which may be written as

$$W = \frac{h - c}{h + \sigma c} \cdot \frac{s}{25{,}050} \cdot Q \, .$$

where Q is taken from line 18. The total heat utilised per pound of steam through the blading is, it will be seen, 227.5 B.Th.U. corresponding to a consumption of about 15 lb. per kilowatt-hour, to which, however, must be added the dummy and gland leakage.

The blade heights, of which approximate values are given in line 11, could be adjusted to any degree of refinement desired, but, as already stated, there will be no appreciable change in the performance of the turbine if they are, in each case, taken at the nearest even dimension.

Detailed Analysis of a Reaction Turbine.

The method given above makes it possible to proportion rationally the blade heights of a turbine from group to group, and it may be surmised that a turbine thus constructed would have the maximum efficiency consistent with its value of λ. There seems, however, to be no very great benefit if "constant angle" blading is used.

It has, in fact, been common to fix the blade heights by simple empirical rules, as described in a previous Chapter. These give very fair results in practice, though with a 28-in. vacuum or more, the steam way at the low-pressure end is liable to be somewhat restricted, and it becomes necessary to fit wing blades here.

We shall analyse, by means of the curves, Figs. 53 to 56, page 98, the turbine, of which the principal proportions were empirically determined in Chapter VII. These main particulars are repeated in the first seven lines of Table VIII., on page 102, whilst the blade speed of each group is set down in line 12.

Using the formulas established in the Chapter on Labyrinth Packings, it appears that the high-pressure dummy leakage is 0.79 lb. per second, that of the intermediate dummy 0.36 lb. per second, and that of the low-pressure dummy about 0.18 lb. per second.

Hence, if 9 lb. of steam flow per second through the low-pressure blading, 8.82 lb. will pass through the intermediate blading and 8.39 lb. through that of the high-pressure section.

In making such an analysis as is proposed, it is by far the most convenient system to start at the exhaust end of the turbine and work backwards, group by group. Small arithmetical errors, and the like, are then non-cumulative.

For a 28-in. vacuum the last row of blades should be fully "winged," but in what follows semi-wing blades have been assumed. The exhaust pressure has, accordingly, been taken as 1.47 lb. absolute, as shown in line 8, and the corresponding specific volume of the steam as 190 cub. ft. per lb. This should be taken at rather a low value so as to avoid getting into the superheated "field" at the high-pressure end of the turbine. To do this has the draw-back that the friction loss with superheated steam is different from that with saturated, a fact which is most conveniently taken into consideration, after the completion of the main calculation, by means of correction curves.

The area through the last row of blades being 2.896 sq. ft., the discharge velocity is $\frac{9 \times 190}{2.896} = 596.8$ ft. per second, so that the ratio δ of blade speed to steam speed is $\frac{275}{596.8} = 0.47$.

With this value of δ the pressure drop required to maintain the flow through a group consisting of six rows of wing blades can be determined by means of the curves in Fig. 54, which have been plotted from the equations established later on in Chapter XIV. on Groups of Blades with Constant Discharge Angle.

Table VIII. Analysis of Reaction Turbine.

		Steam Chest.	High-Pressure Section. Weight through Blades = 8.89 lb. per second.			Intermediate Section. Weight through Blades = 8.82 lb. per second.			Low-Pressure Section. Weight through Blades = 9 lb. per second.					
—	Group Number	—	1	2	3	4	5	6	7	8	9	10	11	12
1	Drum diameter, in.	..	18½	18	18	25½	25½	25½	36	36	36	36	36	36
2	Total rows of blades per group	..	24	24	24	12	12	12	6	6	6	6	6	6
3	Blade height h, in.	..	¾	1	1½	1	1½	2⅛	1½	2⅛	3	4¼	6	6*
4	Clearance c, in.	..	0.020	0.020	0.020	0.025	0.025	0.030	0.050	0.060	0.060	0.070	0.080	0.080
5	$h - c$, in.	..	0.730	0.980	1.480	0.975	1.475	2.095	1.450	2.065	2.940	4.180	5.920	5.920
6	$h + \sigma c$, in.	..	0.793	1.043	1.543	1.054	1.554	2.190	1.607	2.254	3.129	4.400	6.171	6.128
7	Net area through blades Ω,† sq. ft.	..	0.1083	0.1360	0.2062	0.1917	0.2875	0.4150	0.4135	0.5860	0.8375	1.201	1.775	2.896
8	Pressure on discharge from group p_x, lb. per sq. in.	170.2	123.35	85.78	62.84	42.49	29.925	21.135	14.618	9.739	6.428	4.115	2.447	1.47
9	Volume ditto ditto V_x, cub. ft. per lb.	Steam 0.945, dry	3.375	4.695	6.198	8.847	12.168	16.69	24.35	35.27	51.49	73.79	118.6	190.0
10	Velocity v_x, ditto ditto ft. per sec.		274.1	289.6	252.2	407.0	373.3	361.9	529.9	541.7	553.3	553.0	601.3	596.8
11	$p_x V_x$		416.3	402.7	389.4	375.8	364.1	352.7	355.9	343.5	331.0	303.7	290.2	279.3
12	Mean blade speed s, ft. per sec.		124.2	124.2	127.6	173.6	176.6	180.9	245.2	249.4	251.2	263.6	275.0	275.0
13	$\delta = \frac{s}{v_x}$		0.453	0.430	0.506	0.462	0.472	0.500	0.462	0.460	0.455	0.476	0.457	0.467
14	$x = \frac{\text{pressure at entrance to group}}{p_x}$		1.380	1.438	1.366	1.479	1.420	1.416	1.446	1.501	1.515	1.562	1.682	1.665
15	B (From Fig. 56, page 98)		0.856	0.837	0.860	0.825	0.843	0.845	0.835	0.818	0.814	0.800	0.767	0.772
16	$N + 1.05 \log x$		24.147	24.166	24.156	12.179	12.160	12.158	6.168	6.185	6.189	6.203	6.236	6.231
17	v_x B $(N + 1.05 \log x) \cos \alpha$		5377.0	5559.0	4971.8	3881.3	3631.7	3528.5	2728.9	2601.0	2645.5	2604.6	2729.6	2485.5
18	$\frac{Ns}{2}$		1490.4	1490.4	1531.2	1041.6	1059.6	1085.4	735.6	748.2	753.6	790.8	825.0	825.0
19	Q = line 17 − line 18		3886.6	4068.6	3640.6	2839.7	2572.1	2443.1	1993.3	1852.8	1891.9	1813.8	1904.6	1660.5
20	Work done per pound $= \frac{h - c}{h + \sigma c} \cdot \frac{s}{32.2} \cdot Q$, ft.-lb.		13797	14743	13836	14162	13388	13132	13697	13146	13868	14106	15606	13695
21	Ditto ditto B.Th.U.		17.73	18.95	17.78	18.20	17.21	16.88	17.61	16.90	17.82	18.13	20.60	17.61
22	Total heat in 1 lb. of steam on discharge, B.Th.U.	1146.28	1128.55	1109.60	1091.82	1073.62	1056.41	1039.53	1021.92	1005.02	987.20	969.07	948.47	930.86
23	Sensible ditto ditto	339.86	314.50	286.30	264.93	239.90	219.05	199.80	178.20	158.40	139.40	122.70	102.07	83.36
24	Actual latent heat ditto ditto	806.42	814.05	823.30	826.89	833.72	837.36	839.73	843.72	846.62	847.80	846.37	846.40	847.50
25	Latent heat of dry steam at the same pressure, ditto	853.5	872.2	891.8	906.7	924.1	938.7	952.1	967.2	981.0	994.1	1006.0	1020.3	1033.4
26	Volume ditto cub. ft. per lb.	..	3.616	5.085	6.796	9.806	13.64	18.92	27.91	40.87	60.37	87.71	142.9	231.7

* Semi-wing blades. For a 28-in. vacuum the last two rows should be fully "winged."

† Ω = mean circumference $\times (h + \sigma c) \times \sin \alpha \div 144$.

Taking the equation $\phi = \left(\frac{v_N}{100}\right)^2 \frac{N + \mu}{P_N V_N}$ we have $v_N = 596.8$; $p_N V_N = 1.47 \times 190 = 279.3$ and $N = 6$.

As a provisional value, assume $\mu = 0.8$, and we then get

$$\phi = \left(\frac{596.8}{100}\right)^2 \frac{6.8}{279.3} = 0.866.$$

From the curves in Fig. 54 it appears that with $\delta = 0.44$ the corresponding value of x is about 1.63. With this value of x we get 0.74 as a corrected value of μ, giving $\phi = 0.861$, whence the corrected value of $x = 1.665$, so that the pressure at entrance to the group is $1.47 \times 1.665 = 2.447$ lb. per sq. in. absolute.

Having found x, the corresponding value of B is taken from the curve in Fig. 56, its value being 0.772. Then, as shown in Chapter XIV. on the Flow through a Group of Blades, the indicated work done by the group per pound of steam passed per second is

$$\frac{h - c}{h + \sigma c} \cdot \frac{s}{32.2} \cdot \left\{ v_N \,.\, B \,.\, (N + 1.05 \log x) \cos a - \frac{N s^2}{2} \right\} \text{ ft.-lb.}$$

Putting $v_N = 596.8$; $B = 0.772$, and taking $\cos a = 0.8660$ for semi-wing blades, we get

$$v_N \, B \cos a \,.\, (N + 1.05 \log x) \cos a = 2485.5.$$

Subtracting

$$\frac{N s^2}{2} \text{ (line 18)} = \quad 825$$

we get

$$Q = 1660.5,$$

whence

$$\frac{h - c}{h + \sigma c} \cdot \frac{s}{32.2} Q = W = 13{,}695 \text{ ft.-lb. per second per lb. passed,}$$

which is equivalent to 17.61 B.Th.U.

This is the amount of energy which has been removed from each pound of the steam in the shape of useful work as it passed through the group.

Reference to a steam table shows that, as finally discharged from the group, the heat content was 930.86 B.Th.U. per lb., the steam being 0.863 dry, so that before it entered the group its heat content was 930.86 + 17.61 = 948.47 B.Th.U. per lb. Its pressure then was as already found, 2.447 lb. absolute, and at this pressure dry steam has a sensible heat of 102.07 B.Th.U. The actual amount latent is therefore 948.47 − 102.07 = 846.40 B.Th.U. As dry steam at the same pressure has a latent heat of 1020.3 B.Th.U. and a specific volume of 142.9 cub. ft. per lb., the actual

specific volume of the steam on discharge from group 11 is $\frac{142.9 \times 846.40}{1020.3} = 118.6$ cub. ft. per lb. Its velocity of outflow from this group, which consists of normal blades, is therefore $\frac{9 \times 118.6}{1.775} = 601.3$ ft. per second.

This gives

$$\delta = \frac{275}{601.3} = 0.46$$

Whence using the curves for normal blades, Fig. 53, the value of x is found to be 1.682, so that the steam pressure on entering the group is $2.447 \times 1.682 = 4.400$ lb. per sq. in. absolute.

The useful work done by the blades is found as before, cos α being taken as 0.9492, since normal blades have a discharge angle averaging about 18° 20′.

The details of the calculation are shown in the table, and from line 21 it appears that the indicated work in this group amounts to 20.60 B.Th.U. per lb. Hence, before entering the group, the steam had a heat content of $948.47 + 20.60 = 969.07$ B.Th.U. per lb.

Subtracting the sensible heat corresponding to the pressure, the amount actually latent is 846.37, from which the actual specific volume of the steam can be calculated as before, and this is tabulated in line 9, group 10.

The other groups are calculated in succession in the same way, the results being given in the table. Adding the total work done by each group, this, in the case of the low-pressure section, amounts to

$$9\ (17.61 + 16.90 + 17.82 + 18.13 + 20.60 + 17.61) = 977.1 \text{ B.Th.U. per second.}$$

For the intermediate section the indicated work is

$$8.82\ (18.20 + 17.21 + 16.88) = 470.0 \text{ B.Th.U. per second,}$$

and in the case of the high-pressure section the work done is

$$8.39 \times (17.73 + 18.95 + 17.78) = 456.9 \text{ B.Th.U. per second.}$$

The total indicated work done per hour is thus

$$\frac{(977.1 + 470.0 + 456.9)\ 3600}{3412} = 2009 \text{ kw.}$$

The total steam passed per second is

	lb.
Through the low-pressure blading	9.00
„ „ dummy	0.18
„ „ glands (say) ...	0.17
Total	9.35 per second.

Hence the consumption per indicated kilowatt-hour is

$$\frac{3600 \times 9.35}{2009} = 16.75 \text{ lb.}$$

Since, however, the steam was only 0.945 dry, the consumption corrected to dry steam will be 15.83 per indicated kilowatt. This, however, refers to an initial pressure of 170.2 lb. and a 27-in. vacuum. Correcting by Fig. 39, *ante*, to steam at 165 lb., and a 28-in. vacuum gives a consumption of 15.2 lb. per indicated kilowatt-hour. As an ideal turbine would, under these standard conditions, require 10.67 lb. per kilowatt-hour the indicated efficiency ratio is $\frac{10.67}{15.2} = 0.702$, corresponding to a brake efficiency ratio of about 67.4 per cent.

Since Fig. 43 shows about the maximum efficiency attainable with a given value of λ, the above is rather a remarkable result to be reached with an empirically proportioned turbine. The explanation lies in the fact that with such a turbine the true value of λ is considerably in excess of the nominal, since it is the drum diameters which have been increased in the ratio of $\sqrt{2}$ to 1, and not the mean diameters, as theory requires. With the turbine of which the theoretical proportions were got out on pages 95 and 96, an equal efficiency is obtained (see page 115), although the true value of λ is substantially less.

The output with 170 lb. steam and a 27-in. vacuum was found on page 104 to be 2009 indicated kilowatts. Under the standard conditions it will be about $\frac{2009 \times 165}{170} \times 1.043 = 2035$ indicated kilowatts, which is a little on the low side. The vacuum correction of 1.043 is, moreover, a little too great, since Fig. 39 was prepared from the results of tests with turbines having a more liberal steam way at the exhaust end than has been assumed in Table VIII.

CHAPTER XIII.

FLOW OF STEAM THROUGH GROUPS OF BLADES.

Groups with Constant Steam Velocity.

AS has already been pointed out in Chapter VII., axial-flow reaction turbines are commonly divided up into a series of groups, or, as they are popularly termed, "expansions." In each group the blade height is constant, and in many cases each of the constituent rows of blades is identical in form and setting with its predecessor. In other cases the blades are "gauged" so as to keep the velocity of the steam approximately constant from end to end of the group. As the steam passes through a group, it, of course, falls in pressure and expands in volume, and hence, if the speed of flow is to be the same at each row, the net area available for discharge must be progressively increased. This is accomplished by increasing the discharge angle from row to row. Neglecting for the present the effect of clearance over the blade tips, this net area for flow is given by the relation $\Omega = A \sin \theta$, where A denotes the area of the annulus between the rotor and the casing, and θ the discharge angle of the blades. No correction is needed for the fact that the blades have a sensible thickness, the thickness factor for blades of the Parsons type being unity.

In discussing the flow of steam through a reaction turbine each group must be treated as a whole. Two cases arise accordingly, the equations for the flow and for the work done being different when the blades are gauged so as to maintain a constant steam velocity from what they are when it is the discharge angle which is kept constant. The simpler case of the two is when the blades are gauged to give a constant steam velocity.

Theory shows that the speed of inflow into a group is not independent of the blade speed, the velocity of the steam increasing

slightly the slower the speed of the blade. If we let $\delta = \frac{\text{blade speed}}{\text{steam speed}}$, then, as is proved later, the speed of inflow into a group of blades, gauged so as to keep the steam speed constant, is given approximately by the following relations:—

δ	Steam velocity v. ft. per second.	
0.4	... ... $238\sqrt{\frac{U_g}{N}}$	(23)
0.5	... ... $231\sqrt{\frac{U_g}{N}}$	(24)
0.6	... ... $226\sqrt{\frac{U_g}{N}}$	(25)

In the above N denotes the total number of rows, both fixed and moving, in the group, whilst U_g denotes the heat accounted for in the group. If the turbine is so proportioned that an equal quantity of work is done in each group, and if u be the total heat which would be converted into work in a perfect turbine with the same admission and condenser pressure, then $U_g = \frac{u}{G} \times$ reheat factor, where G denotes the total number of groups. An approximate estimate of the probable indicated efficiency ratio of a proposed turbine can always be obtained by reference to Fig. 43, page 50, and thence the reheat factor from Table III. For many purposes, however, it is sufficient to replace U_g by u_g, where u_g denotes the theoretical available heat corresponding to the pressure drop in the group, as measured direct from the Mollier diagram.

When $\frac{p_0}{p_n}$, *i.e.*, the absolute pressure in front of a group divided by the pressure behind the group, is of the order of $\sqrt{2}$, the corresponding efficiencies are approximately as follows:—

δ	Group efficiency.	
0.4	... ... $0.653\,\frac{h - c}{h + 1.8\,c}$	(26)
0.5	... ... $0.717\,\frac{h - c}{h + 1.8\,c}$	(27)
0.6	... ... $0.763\,\frac{h - c}{h + 1.8\,c}$,	(28)

where h denotes the blade height and c the tip clearance, both in inches It is assumed in the above that the first row of blades is gauged to about one-third the pitch.

TABLE IX.—PROPERTIES OF STEAM ON EXPANSION FROM AN INITIAL PRESSURE OF 154.8 LB. PER SQUARE INCH ABSOLUTE WHEN THREE-TENTHS OF THE HEAT AVAILABLE AT EACH STAGE IS WASTED IN FRICTION.

Available Heat per Pound.	Volume of 1 lb. after Expansion.	Increment of Volume per Unit of Available Heat.	Absolute Pressure.	Available Heat per Pound.	Volume of 1 lb. after Expansion.	Increment of Volume per Unit of Available Heat.	Absolute Pressure.
B.Th.U.	cub. ft.	cub. ft.	lb. per sq. in.	B.Th.U.	cub. ft.	cub. ft.	lb. per sq. in.
0	2.878	0.0328	154.8	190	29.55	0.414	11.90
5	3.042	0.0346	145.6	195	31.62	0.436	11.04
10	3.215	0.0366	137.0	200	33.80	0.482	10.23
15	3.398	0.0388	128.7	205	36.21	0.502	9.506
20	3.592	0.0416	121.1	210	38.72	0.530	8.835
25	3.800	0.0440	113.8	215	41.37	0.568	8.206
30	4.020	0.0466	106.9	220	44.21	0.610	7.626
35	4.253	0.0496	100.4	225	47.26	0.674	7.076
40	4.501	0.0530	94.32	230	50.63	0.742	6.560
45	4.766	0.0562	88.55	235	54.34	0.820	6.074
50	5.047	0.0598	83.12	240	58.44	0.921	5.619
55	5.346	0.0636	78.00	245	63.06	1.002	5.194
60	5.664	0.0686	73.16	250	68.07	1.094	4.793
65	6 007	0.0728	68.61	255	73.54	1.200	4.416
70	6 371	0.0764	64.31	260	79.54	1.324	4.057
75	6.753	0.0820	60.27	265	86.16	1.440	3.712
80	7.163	0.0882	56.45	270	93.36	1.568	3.388
85	7.604	0.0932	52.86	275	101.2	1.78	3.088
90	8.072	0.0998	49.49	280	110.1	1.92	2.814
95	8.571	0.1066	46.31	285	119.7	2.12	2.567
100	9.104	0.1148	43.32	290	130.3	2.28	2.343
105	9.678	0.1224	40.51	295	141.7	2.52	2.136
110	10.29	0.130	37.85	300	154.3	2.78	1.947
115	10.94	0.142	35.36	305	168.2	3.04	1.774
120	11.65	0.152	33.01	310	183.4	3.10	1.614
125	12.41	0.164	30.79	315	199.9	3.62	1.470
130	13.23	0.174	28.71	320	218.0	3.96	1.337
135	14.10	0.188	26.75	325	237.8	4.36	1.119
140	15.04	0.200	24.92	330	259.6	4.74	1.106
145	16.04	0.216	23.19	335	283.3	5.20	1.006
150	17.12	0.234	21.58	340	309.3	5.54	0.9143
155	18.29	0.252	20.07	345	337.0	6.20	0.8324
160	19.55	0.270	18.66	350	368.0	6.74	0.7575
165	20.90	0.294	17.32	355	401.7	7.48	0.6892
170	22.37	0.316	16.06	360	439.1	8.34	0.6257
175	23.95	0.342	14.91	365	480.8	9.32	0.5540
180	25.66	0.382	13.85	370	527.4	...	0.5128
185	27.57	0.396	12.83				

Table IX., above, is sometimes useful for fixing the gauging required to maintain a constant steam velocity throughout a group. It is based on the assumption that the losses by friction, &c., at any row, are equal to 30 per cent. of the energy liberated by the fall of pressure through the row. Considerable differences in

the amount of these losses do not much affect the volume corresponding to a given expenditure of available heat.

The expressions given on page 107 for the steam velocity are approximate only. The complete equation for the flow is

$$\left(\frac{v}{224}\right)^2 = \frac{m\,U_g}{N\left\{1 - M\left[1 + \delta^2 - \delta\left(\cos\alpha_0 + \cos\alpha_n\right)\right]\right\} + M\left(\cos\alpha_0 - \cos\alpha_n\right)},$$

where m and M are coefficients, which for the usual velocities of flow may be taken as 0.89 and 0.50 respectively, δ the ratio of blade speed to steam speed, α_0 the angle of discharge which would give the required area of flow with the steam at its initial specific volume, and α_n the discharge angle for the last row of blades.

This equation may be established as follows:—

At the n^{th} row of a group the energy which maintains the flow through it consists of two parts, viz., the kinetic energy carried over with the steam from the previous row, and, secondly, the energy liberated by the expansion of the steam as it passes through the blades. The carry-over of kinetic energy is equal to $\frac{r^2_{n-1}}{2g}$ ft.-lb., where r denotes the velocity of the entering steam measured relatively to the row under discussion. This energy may also be expressed as $\left[\frac{r_{n-1}}{224}\right]^2$ B.Th.U.

The energy liberated by the expansion of the steam as it passes through the n^{th} row may be denoted by q_n B.Th.U., so that the total supply to the row is $\left(\frac{r_{n-1}}{224}\right)^2 + q_n$, and this generates the velocity v with which the steam is discharged from the row. The energy corresponding to the velocity v is $\left(\frac{v}{224}\right)^2$ B.Th.U., and this in the case of gauged blades is the same for all rows. Owing to frictional losses, &c., $\left(\frac{v}{224}\right)^2$ is always less than the energy supplied to the row, and we may therefore write

$$M\left(\frac{r_{n-1}}{224}\right)^2 + m\,q_n = \left(\frac{v}{224}\right)^2,$$

where M and m are coefficients, the values of which have been determined from the results obtained on the test of a large marine turbine. The steam speeds here were of the order of

200 ft. to 300 ft. per second, and an application of the method of least squares to the trial data gave $M = 0.5230$ and $m = 0.9014$, with a very small probable error. As, on the whole, it is better to take coefficients too small rather than too large, the values $M = 0.50$ and $m = 0.89$ have, as already stated, been adopted by the author.

For the $(n+1)^{\text{th}}$ row we have a similar equation to that just found, viz.,

$$M\left(\frac{r_n}{224}\right)^2 + m\, q_{n+1} = \left(\frac{v}{224}\right)^2.$$

Subtracting we get

$$m\,(q_{n+1} - q_n) = -\, M \, . \, \frac{r_n^2 - r_{n-1}^2}{224^2},$$

which, with the notation of the calculus of finite differences, may be written in the form

$$m\,\Delta\, q_n = -\, \frac{M}{224^2} \, . \, \Delta\, r_{n-1}^2,$$

$\Delta\, q_n$ denoting $q_{n+1} - q_n$, and $\Delta\, r_{n-1}^2$ denoting $r_n^2 - r_{n-1}^2$.

Now, if q_1 be the energy liberated by expansion in row 1, and q_2 that liberated on expansion at row 2, and so on, we have

$$\begin{aligned} q_2 &= q_1 + \Delta\, q_1 \\ q_3 &= q_2 + \Delta\, q_2 = q_1 + \Delta\, q_1 + \Delta\, q_2 \\ &\cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \\ &\cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \\ q_n &= q_1 + \Delta\, q_1 + \Delta\, q_2 + \text{\&c.} + \Delta\, q_{n-1}. \end{aligned}$$

Now, the discharge velocity is the same at the first and the final rows, so that q_n cannot differ greatly from q_1, and the curve obtained on plotting q against n must, of necessity, be very flat. Hence no appreciable error will be made if each of the values $\Delta\, q_1$, $\Delta\, q_2$, &c., is made equal to a mean value $\Delta\, q$, say, where

$$\Delta\, q = \frac{\Delta\, q_1 + \Delta\, q_2 + \text{\&c.} + \Delta\, q_n}{N}.$$

From the relation already found between $\Delta\, q_n$ and $\Delta\, r_{n-1}^2$ we have therefore

$$\Delta\, q = -\frac{1}{N} \cdot \frac{M}{m} \cdot \left(\frac{1}{224}\right)^2 \{\Delta\, r_0^2 + \Delta\, r_1^2 + \Delta\, r_2^2 + \text{\&c.} + \Delta\, r_{n-1}^2\}.$$

Now, from Fig. 49, Chapter X., it will be seen that

$$r^2 = v^2 + s^2 - 2\, v\, s \cos \alpha,$$

where s denotes the blade speed.

Hence

$$\Delta r_0^2 = r_1^2 - r_0^2 = 2\,v\,s\,(\cos\alpha_0 - \cos\alpha_1)$$
$$\Delta r_1^2 = r_2^2 - r_1^2 = 2\,v\,s\,(\cos\alpha_1 - \cos\alpha_2).$$
$$\cdots\cdots\cdots\cdots$$
$$\Delta r_{N-1}^2 = r_N^2 - r_{N-1}^2 = 2\,v\,s\,(\cos\alpha_{N-1} - \cos\alpha_N).$$

Hence

$$\Delta r_0^2 + \Delta r_1^2 + \Delta r_2^2 + \text{\&c.} = 2\,v\,s\,[\cos\alpha_0 - \cos\alpha_N].$$

Here α_0 denotes the value of the discharge angle which would give the requisite area of flow for the steam in its condition in front of the first row of blades, when it has the specific volume V_0, and α_N denotes the discharge angle for the last row of blades when the specific volume is V_N.

The following relation subsists between α_0 and α_N, viz.,

$$\frac{V_N}{V_0} = \frac{\sin\alpha_N}{\sin\alpha_0}.$$

In general V_0 and V_N are both known, and if α_0 be known or assumed, α_N can be found immediately.

We thus get

$$\Delta q = -\,.\,\frac{M}{m}\left(\frac{1}{224}\right)^2 .\,2\,v\,s\,\frac{[\cos\alpha_0 - \cos\alpha_N]}{N\,.},$$

and if we put $s = \delta\,v$, this becomes

$$\Delta q = -\frac{M}{m}\left(\frac{v}{224}\right)^2 .\,2\,\delta\left\{\frac{\cos\alpha_0 - \cos\alpha_N}{N}\right\}.$$

We have the following values for the heat expenditure at the different rows:—

Row Number.	Heat Expenditure.
1	q_1
2	$q + \Delta q$
3	$q + 2\,\Delta q$
. . .	. . .
. . .	. . .
N	$q + (N - 1)\,\Delta q.$

The successive values form an arithmetical progression, so that U_g, the total heat expenditure in the N rows, is

$$U_g = U_n - U_0 = N\,q_1 + \frac{N\,.\,(N - 1)}{2}\,\Delta q.$$

Substituting the value already found for Δq we get

$$U_g = U_n - U_0 = N\,q_1 - \frac{M}{m}\,(N - 1)\left(\frac{v}{224}\right)^2 .\,\delta\,(\cos\alpha_0 - \cos\alpha_N).$$

If we assume the existence of an imaginary Row, No. 0, before

Row No. 1, we shall make no appreciable error in our deductions whilst making the mathematics more symmetrical.

In that case we have

$$M\left[\frac{r_0}{224}\right]^2 + m\,q_1 = \left(\frac{v}{224}\right)^2.$$

But, from Fig. 49, Chapter X., *ante*, we have if $\delta = \frac{s}{v}$;

$$r_0^2 = v^2\,(1 + \delta^2 - 2\,\delta\cos\alpha_0).$$

Hence

$$q_1 = \left(\frac{v}{224}\right)^2\left\{\frac{1}{m} - \frac{M}{m}\,.\,[1 + \delta^2 - 2\,\delta\cos\alpha_0]\right\}.$$

Substituting this we get

$$U_g = \left(\frac{v}{224}\right)^2\left[\frac{N}{m} - \frac{NM}{m}\left[\,.\,1 + \delta^2 - \delta\,(\cos\alpha_0 + \cos\alpha_N)\right] + \frac{M}{m}\,.\,\delta\,(\cos\alpha_0 - \cos\alpha_N)\right].$$

The last term on the right-hand side may, in practice, be neglected, in which case we get

$$\left(\frac{v}{224}\right)^2 = \frac{U}{N\left[\frac{1}{m} - \frac{M}{m}\,.\,\left\{1 + \delta^2 - \delta\,(\cos\alpha_0 + \cos\alpha_N)\right\}\right]} \quad . \quad (29)$$

Taking $m = 0.89$ and $M = 0.50$, and putting $\cos\alpha_0 + \cos\alpha_N$ as equal to 1.85, which is a fair average value, we get the values of v already given on page 107, equations (23) to (25).

As an example we may take the case of steam supplied at a pressure of 154.8 lb. absolute before the first row of blades, and that it is intended that the steam speed in the first group is to be 240 ft. per second, the blade speed being 120 ft. per second. Then $\delta = \frac{120}{240} = 0.5$

Hence
$$\frac{U}{N} = \left[\frac{240}{231}\right]^2 = 1.039.$$

If the number of rows in the group be 26, we get

$$U = 26 \times 1.039 \times 26 = 27.01 \text{ B.Th.U.}$$

as the heat expenditure in the first group.

From Table IX. it appears accordingly that the volume on discharge from the group will be 3.878 cub. ft. per lb., the corresponding pressure being 111.0 lb. per sq. in. absolute. If the angle of discharge for the first row of blades is taken as 18 deg., the angle α_N of discharge at the last row will be given by the relation

$$\sin\alpha_N = \frac{3.878}{2.878}\,.\,\sin 18 \text{ deg.} = 0.4163,$$

so that $\alpha_N = 24$ deg. 36 min.

Strictly speaking, a correction should be made for the effects of tip clearance, but the refinement is really unnecessary.

If the blade height be 1 in., and the clearance 20 mils, the efficiency of the blading will be $0.717 \times \frac{1 - 0.020}{1 + 0.036} = 0.678$, or, in other words, the indicated work done by the first group will be $27.01 \times 0.678 = 18.31$ B.Th.U. per pound of steam passed.

The formulas above given for the efficiency are derived as follows:—

Neglecting, for the present, the effects of tip clearance, the work done by a pair of rows, per pound of steam passed per second, is (as shown in Chapter X., page 80, *ante*)

$$\frac{s}{g}\left[v \cos \alpha_1 + v \cos \alpha_2 - s\right] \text{ ft.-lb.}$$

For the whole group therefore

$$W = \frac{s}{g} \cdot \left[v\, \Sigma_1^N \cos \alpha - \frac{N\, s}{2}\right].$$

Now $\cos \alpha$ gives so flat a curve when plotted against n that we may write

$$\Sigma_1^N \cos \alpha = \frac{N}{2} \cdot \{ \cos \alpha_0 + \cos \alpha_N \}.$$

Whence

$$W = \frac{s\, N}{2\, g} \cdot \left[v\, (\cos \alpha_0 + \cos \alpha_N) - s\right].$$

Putting $s = \delta \,.\, v$ this becomes

$$W = \frac{N\, v^2}{2\, g} \cdot \left[\delta\, (\cos \alpha_0 + \cos \alpha_N) - \delta^2\right] \text{ft.-lb.}$$

$$= \frac{N \,.\, v^2}{224^2} \cdot \left[\delta\, (\cos \alpha_0 + \cos \alpha_N) - \delta^2\right] \text{ B.Th.U.}$$

But

$$\frac{v^2}{(224)^2} = \frac{U}{N\left[\frac{1}{m} - \frac{M}{m}\left\{1 + \delta^2 - \delta\, (\cos \alpha_0 + \cos \alpha_N)\right\}\right]}.$$

Hence the efficiency $\frac{W}{U}$ is

$$\frac{\delta\, (\cos \alpha_0 + \cos \alpha_N) - \delta^2}{\frac{1}{m} - \frac{M}{m} \cdot \left[1 + \delta^2 - \delta\, (\cos \alpha_0 + \cos \alpha_N)\right]} \quad . \quad . \quad . \quad (30)$$

Taking the values already given for $\cos \alpha_0 + \cos \alpha_N$ and for m and M, and allowing for the losses over blade tips, this expression reduces to the equations already given for the efficiency on page 107, *ante*.

The equations above established may be used to determine the output and the efficiency of the turbine of which the blading was got out on page 95. The initial conditions were a supply of dry steam at 165 lb. absolute, whilst the exhaust pressure was equivalent to a 28-in. vacuum. Hence the theoretical available heat is 321 B.Th.U. Allowing for a reheat factor of 1.067, the heat which actually becomes available is 342 B.Th.U., and the blading was proportioned so that this total was equally divided amongst the twelve groups constituting the turbine. Hence 28.5 heat units will be expended in each group. With the turbine running at 1500 revolutions per minute it will be found on making a trial calculation of the steam speed by means of equations (23) to (25), page 107, that the ratio $\delta = \dfrac{\text{blade speed}}{\text{steam speed}}$ is practically constant throughout and equal to 0.5.

Taking Group No. 1, the specific volume of the steam in front of the group is 2.757, whilst on discharge its specific volume is (from Table VI., page 94) 3.780. Hence, if the discharge angle of the first row of blades is 18 deg., that of the last row will be given by

$$\sin \alpha_N = \frac{\sin 18^\circ \times 3.780}{2.757} = 0.4240$$

so that α_N is about 25° 10′, and $\cos \alpha_N$ is 0.9050.

Hence

$$\cos \alpha_0 + \cos \alpha_N = 1.856.$$

From formula (29) we have

$$\left(\frac{v}{224}\right)^2 = \frac{0.89\ \text{U}}{\text{N}\,.\,[1 - 0.5\,\{1 + \delta^2 - \delta\,(\cos \alpha_0 + \cos \alpha_N)\}]}.$$

Hence $v = 251.5$ ft. per second, if $\delta = 0.5$.

Actually

$$s = \frac{19 \times \pi \times 1500}{12 \times 60} = 124.7 \text{ ft. per second,}$$

so the true value of δ is 0.496, which is near enough to the assumed value as to necessitate no correction of v.

If the tip clearance is 20 mils, the blade height being $\frac{3}{4}$ in., the efficiency of the group is from formula (27)

$$\frac{0.75 - 0.020}{0.75 + 1.8 \times 0.020} \times 0.717 = 0.668.$$

In the case of the last group of the turbine at the low-pressure end we note from Table VI., page 94, that the specific

volume of the steam in front of the group is 180.3 cub. ft. per lb., whilst the volume on discharge is 292.4. Hence, taking as before 18 deg. as the initial blade angle we have

$$\sin \alpha_N = \frac{\sin 18^\circ \times 292.4}{180.3} = 0.501 .$$

so that α_N is 30 deg. nearly.

Hence $\cos \alpha_0 + \cos \alpha_N = 1.817$. This is slightly less than the value assumed in deducing formulas (24) and (27). Taking δ as 0.5 we get, however,

$$\left(\frac{v}{224}\right)^2 = \frac{0.89 \times 28.5}{4\left[1 - 0.5\left(1 + 0.25 - \frac{1}{2} \times 1.817\right)\right]}$$
$$= \frac{0.89 \times 28.5}{4 \times 0.830} .$$

Hence $v = 620$ ft. per second, and as $s = 309$ ft. per second, the assumed value of δ is again practically correct.

From equation (30), page 113, the efficiency is equal to

$$\frac{h - c}{h + \sigma c} \cdot \frac{0.89\left[\delta\left(\cos \alpha_0 + \cos \alpha_N\right) - \delta^2\right]}{1 - 0.5\left[1 + \delta^2 - \delta\left(\cos \alpha_0 + \cos \alpha_N\right)\right]}$$

Taking c as 0.085 in., and substituting the value above found for $\cos \alpha_0 + \cos \alpha_N$. this gives the efficiency of the group as 0.687.

If the efficiency of the other groups is calculated in the same way, we get the following values:—

Group Number.	Efficiency.	Group Number.	Efficiency.
1	0.668	7	0.675
2	0 669	8	0.683
3	0.681	9	0.689
4	0.675	10	0.691
5	0.679	11	0.681
6	0.705	12	0.687

The mean value is thus 0.682, so that, apart from the losses due to gland and dummy leakage, the efficiency ratio of the proposed turbine would be 0.726. These leakage losses can be approximately calculated, as already explained. Taking them as 4 per cent., the "indicated" efficiency ratio becomes 0.696, which, with a mechanical efficiency of 0.97, gives 0.676 as the brake efficiency ratio, a figure which is in good agreement with that obtained from the curve, Fig. 43, page 50.

CHAPTER XIV.

GROUPS OF BLADES WITH CONSTANT DISCHARGE ANGLE.

THE problem of the flow of steam through a group in which the rows of blades are all gauged to the same opening is complicated and results in equations so cumbrous that the results are best expressed by tables and curves, such as have already been given in Chapters XI. and XII.

The curves there used in computing the velocity of flow into, and the work done by, groups of ungauged blades have been plotted from equations deduced in the following way:—It has long been known that over a very considerable range the relation between the volume and pressure of 1 lb. of steam can be expressed with substantial accuracy by an equation of the form

$$P V^{\gamma} = \text{constant}.$$

In the case of the group of a reaction turbine the ratio of expansion is never more than 2.5 at most, and is more commonly of the order of 1.29 to 1.70. Within such limits the law in question may be taken not merely as approximately accurate but as absolutely true.

The value of the index γ, when steam expands through a group of turbine blades, depends upon the efficiency of this group. The curve in Fig. 57 shows how, in the case of initially dry steam, γ varies with different values of η; the three sets of points plotted correspond respectively to cases in which the ratio of the initial to the final pressure is respectively 9, 81, and 243. It will be seen

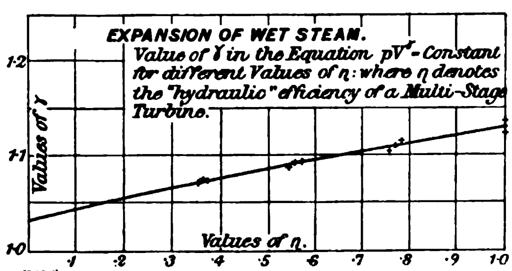


Fig. 57.

that for such efficiencies as are usual in turbine practice the index has but a small range of values. As one consequence of this, it

turns out, as will be proved in the sequel, that the velocity of flow of saturated steam through a group is almost independent in practice of the actual law of the expansion within the group.

The curve in question has been drawn for the case of steam initially dry. The index is however, to a certain extent, dependent upon the initial wetness of the steam, but any variation due to this is covered by the fact just stated.

Premising that the law of expansion can be expressed as above set forth, consider a group consisting of n stages, half of which are constituted by the fixed rows and half by the moving rows. Let $v_1, v_2, v_3 \ldots v_n$ be the velocities of outflow of the steam from the corresponding stages.

Referring to Fig. 50, page 79, let A B = v_1 denote the direction and magnitude of the velocity of the steam as it issues from the first row of guide blades, and let C B = s denote the speed of the blading. The momentum of each pound of steam as it issues is equal to $\frac{v_1}{g}$, and the component of this momentum in the direction of rotation is G B $\div g$, or

$$\frac{v_1 \cos a}{g}.$$

Taking now stage No. 2, which is a row of moving blades, the steam issues from this with a velocity which, measured relatively to the moving blades, is given by the line H I = v_2. This, as stated, is a relative velocity, and the absolute velocity, in space, of the steam as it issues is obtained by setting off I J = s and joining H J. The absolute momentum of 1 lb. of the steam as it issues is therefore equal to H J $\div g$. The momentum of this in the direction of the motion of the blades is $-$ K J $\div g$, which, it will be seen, is equal to

$$-\frac{v_2 \cos a - s}{g}.$$

This component is directed in the opposite direction to G B, and hence the total change of momentum parallel to the direction of its own motion, which the moving blades have effected on the pound of steam is

$$\frac{\text{G B} + \text{K J}}{g} = \frac{v_1 \cos a + v_2 \cos a - s}{g}.$$

By an elementary principle in mechanics, the force acting on a body is equal to the change of momentum effected per

second in the same direction. Hence, if w pounds of steam pass through stage No. 2 per second, the force exerted on this mass in the plane of rotation is equal to

$$\frac{w}{g}\left\{(v_1 + v_2)\cos\alpha - s\right\}.$$

Since action and reaction are equal and opposite, this is also the force tending to rotate the wheel.

The total rotating force exerted on a group is therefore the sum of a similar quantity to the above for every row of moving blades. This may be called the impulse on the group, and this impulse is therefore equal to

$$\frac{w}{g}\cdot\left\{\cos\alpha\,\Sigma_1^N v - \frac{N s}{2}\right\},$$

where N denotes the total number of stages, which is, of course, twice the number of moving rows in the group.

In passing through the various stages of a group the steam expands, and its specific volume V increases. If Ω be the area of the passage-way, the velocity v is given by the relation $V w = v\,\Omega$, so that the expression just given may be written

$$\frac{w}{g}\left\{\frac{w}{\Omega}\cos\alpha\,\Sigma_1^N V - \frac{N s}{2}\right\},$$

where $\Sigma_1^N V$ denotes the sum of the specific volumes of the steam measured at each successive stage.

In general V_0, the initial volume of the steam, is known, as are also p_0 and p_N, where p_N denotes the pressure on discharge from the group.

If the law by which V increases stage by stage from V_0 to V_N were known, the value of $\Sigma_1^N V$ could be readily found. The simplest assumption, no doubt, is that the value of V plotted against n gives a straight line, but this over-estimates the work done, since the true curve is convex towards the axis of n.

A more accurate assumption is that

$$V_n = V_0\,\mu^n,$$

where μ can be obtained from the relation that $V_N = V_0\,\mu^N$, or $N \log \mu = \log \rho_N$

This makes

$$\sum_1^N V = V_0 \frac{\rho_N - 1}{1 - \left(\frac{1}{\rho_N}\right)^{\frac{1}{N}}}$$

whence the impulse on the group is

$$\frac{v_0}{g}\left\{\cos\alpha \frac{\rho_{\mathrm{N}} - 1}{1 - \left(\frac{1}{\rho_{\mathrm{N}}}\right)^{\frac{1}{\mathrm{N}}}} - \frac{\mathrm{N}\,s}{2}\right\}.$$

This, though a better approximation to the truth than the assumption that the relation between V and n is expressed by a straight line law, is still not satisfactory, and again over-estimates the work done; because the assumed law $\mathrm{V} = \mathrm{V}_0\,\mu^n$ makes the ratio of expansion the same for each successive stage. Since the volume of the steam increases, however, from row to row, more work is necessarily expended at each successive stage in forcing a given weight of steam through it, and thus actually the ratio of expansion must increase from stage to stage, so as to provide for the additional energy required.

An approximation to the true law of increase of volume can be established as follows:—It appears to be a reasonable assumption that the kinetic energy generated at each stage shall be approximately proportional to the expansion energy* expended in the stage.

The drop of pressure at each stage is small, and calling it $\Delta\,\mathrm{P}$, the expansion energy accounted for in the stage, per pound of steam passed, may be taken as equal to $\mathrm{V}\,\Delta\,\mathrm{P}$.

The kinetic energy per pound of steam leaving the stage is, of course, equal to $\dfrac{v^2}{2g}$.

Thus, on the assumption above made, we may write

$$\mathrm{V}\,\Delta\,\mathrm{P} = \lambda\,.\,v^2,$$

where λ is some constant, but v^2 is proportional to V^2; hence we may write

$$\mathrm{V}\,\Delta\,\mathrm{P} = \beta\,\mathrm{V}^2,$$

or

$$\Delta\,\mathrm{P} = \beta\,\mathrm{V},$$

where β is another constant.

Assuming the relation between P and V to be $\mathrm{P\,V}^\gamma$ = constant, we thus get $\Delta\,\mathrm{P} = l\,\mathrm{P}^{-\frac{1}{\gamma}}$, where l is some constant to be determined later on.

Now $\Delta\,\mathrm{P}$ is the decrease in pressure for a unit increase in

* This may also be called the "available heat" expended in the stage.

the number of stages, and the solution of the above equation is accordingly a problem in the Calculus of Finite Differences.

An approximate solution can be obtained as follows:—

Obviously, if P_0 be the initial pressure, and P the pressure behind some stage N, we must have

$$P_0 - P = \Sigma_1^N \Delta P$$

the total pressure difference being the sum of the pressure differences at each intervening stage.

We may write, therefore,

$$P_0 - P = \Sigma_1^N \Delta P = l \Sigma_1^N P^{-\frac{1}{\gamma}}. \quad . \quad . \quad . \quad (31)$$

Suppose the ordinates in Fig. 58 represent successive values of $P^{-\frac{1}{\gamma}}$, then $\Sigma_1^N P^{-\frac{1}{\gamma}}$ is equal to the area of the dotted rectangles shown. This area is, in its turn, practically equal to that of the curve A E G C, less the rectangle A B C D, plus the rectangle E F G H.

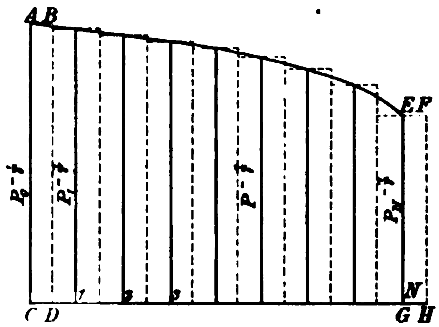


Fig. 58.

Since

$$A\,C = P_0^{-\frac{1}{\gamma}}, \text{ and } E\,G = P^{-\frac{1}{\gamma}},$$

the rectangle A D is numerically equal to $\frac{1}{2} P_0^{-\frac{1}{\gamma}}$, and the rectangle E H to $\frac{1}{2} P^{-\frac{1}{\gamma}}$.

Hence from (31) we get

$$P_0 - P = l\left(\text{area } A\,E\,G\,C + \tfrac{1}{2} P^{-\frac{1}{\gamma}} - \tfrac{1}{2} P_0^{-\frac{1}{\gamma}}\right). = l\int P^{-\frac{1}{\gamma}} d\,n + \frac{l}{2} P^{-\frac{1}{\gamma}} - \frac{l}{2} P_0^{-\frac{1}{\gamma}}.$$

Differentiating with regard to n we get (since P_0 is constant)

$$-\frac{d\,P}{d\,n} = l\,P^{-\frac{1}{\gamma}} - \frac{l}{2\gamma} . P^{-\frac{1}{\gamma}-1} . \frac{d\,P}{d\,n}.$$

Multiplying by $P^{\frac{1}{\gamma}}$ this equation may be written

$$-P^{\frac{1}{\gamma}}\frac{d\,P}{d\,n} = l - \frac{l}{2\gamma} . P^{-1} . \frac{d\,P}{d\,n}.$$

On integration this gives

$$-\frac{\gamma}{\gamma+1} . P^{\frac{\gamma+1}{\gamma}} = l\,n - \frac{l}{2\gamma} . \log_\epsilon P + A \quad . \quad . \quad . \quad (32)$$

where A is the constant of integration.

To determine l and A, we note that when $n = 0$, $P = P_0$, and when $n = N$; $P = P_N$, P_0 and P_N being respectively the initial and final pressures of the steam.

We thus have

$$-\frac{\gamma}{\gamma+1} \cdot P_0^{\frac{\gamma+1}{\gamma}} = -\frac{l}{2\gamma} \cdot \log_e P_0 + A \quad . \quad . \quad . \quad (33)$$

and

$$-\frac{\gamma}{\gamma+1} \cdot P_N^{\frac{\gamma+1}{\gamma}} = lN - \frac{l}{2\gamma} \cdot \log_e P_N + A.$$

Subtracting we get

$$l\left[N + \frac{1}{2\gamma}(\log_e P_0 - \log_e P_N)\right] = \frac{\gamma}{\gamma+1} \cdot \left\{P_0^{\frac{\gamma+1}{\gamma}} - P_N^{\frac{\gamma+1}{\gamma}}\right\},$$

which if $\frac{P_0}{P_N} = x_N$ may be written as

$$l\left(N + \frac{1}{2\gamma}\log_e x\right) = \frac{\gamma}{\gamma+1} \cdot P_0^{\frac{\gamma+1}{\gamma}}\left\{1 - \left(\frac{1}{x_N}\right)^{\frac{\gamma+1}{\gamma}}\right\}.$$

But if ρ_N be the ratio of the volumes at the beginning and end of the expansion, we have

$$\rho_N = x_N^{\frac{1}{\gamma}}$$

and thus we get

$$l = \frac{\gamma}{\gamma+1} \cdot P_0^{\frac{\gamma+1}{\gamma}}\left(1 - \frac{1}{\rho_N x_N}\right) \div \left\{N + \tfrac{1}{2}\log_e \rho_N\right\}.$$

Substituting this value for l in equation (33) we get

$$A = \frac{\gamma}{\gamma+1} \cdot P_0^{\frac{\gamma+1}{\gamma}}\left\{\frac{\left(1 - \frac{1}{\rho_N x_N}\right)}{N + \frac{1}{2}\log_e \rho_N} \frac{\log_e P_0}{2\gamma} - 1\right\}.$$

Hence, substituting in equation (32) the values found for A and l, and letting P denote the pressure after any stage n, and if also $\frac{P}{P_0} = x$, whilst $\frac{V}{V_0} = \rho$, the following relation subsists between n, x, and ρ.

$$\frac{n + \frac{1}{2}\log_e \rho}{N + \frac{1}{2}\log_e \rho_N} = \frac{1 - \frac{1}{x\rho}}{1 - \frac{1}{x_N \rho_N}},$$

which, since $\rho = x^{\frac{1}{\gamma}}$, may also be written

$$\frac{n + \frac{1}{2}\log_e \rho}{N + \frac{1}{2}\log_e \rho_N} = \frac{1 - \left(\frac{1}{\rho}\right)^{1+\gamma}}{1 - \frac{1}{x_N \rho_N}} \quad . \quad . \quad . \quad . \quad (34)$$

The way the volume increases from stage to stage within the group is well shown in Fig. 59, where the ratio of expansion has been plotted against the number of the row.

For many purposes the logarithmic terms on the left may be omitted, and we may write simply

$$\frac{n}{\mathrm{N}} = \frac{1 - \left(\frac{1}{\rho}\right)^{1+\gamma}}{1 - \frac{1}{x_{\mathrm{N}}\,\rho_{\mathrm{N}}}}.$$

The object at issue in deducing equation (34) has been to evaluate

$$\Sigma^{\mathrm{N}} \mathrm{V}.$$

Since $\mathrm{V} = \rho\,\mathrm{V}_0$, we may write

$$\sum_1^{\mathrm{N}} \mathrm{V} = \mathrm{V}_0 \sum_1^{\mathrm{N}} \rho.$$

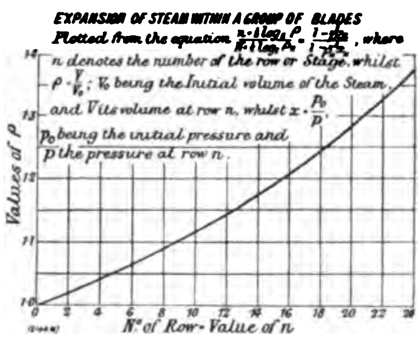


Fig. 59.

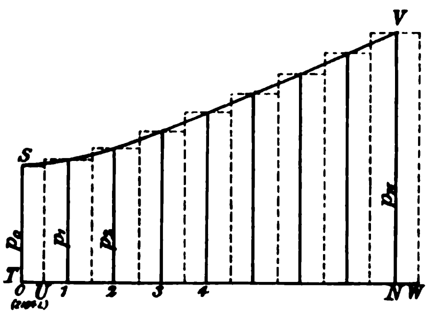


Fig. 60.

Suppose the successive values of ρ are plotted down against n, as in Fig. 60, then the value of $\sum_1^{\mathrm{N}} \rho$ is equal to the area of all the dotted rectangles, which, it will be seen, is practically equivalent to the area of the curve S T V N, minus the rectangle S U, and plus the rectangle V W. Hence we may write

$$\sum_1^{\mathrm{N}} \rho = \int_0^{\mathrm{N}} \rho\, d\, n + \tfrac{1}{2}(\rho_{\mathrm{N}} - \rho_0).$$

To find $\int \rho\, d\, n$, differentiate equation (34), *supra*, with respect to n. This gives

$$d\, n + \frac{1}{2\,\rho} . \, d\,\rho = \frac{\mathrm{N} + \frac{1}{2}\log_\epsilon \rho_{\mathrm{N}}}{1 - \frac{1}{\rho_{\mathrm{N}}\,x_{\mathrm{N}}}} . \, (\gamma + 1)\,\rho^{-\gamma-2} . \, d\,\rho.$$

Whence

$$\rho\, d\, n = \frac{\mathrm{N} + \frac{1}{2}\log_\epsilon \rho_{\mathrm{N}}}{1 - \frac{1}{\rho_{\mathrm{N}}\,x_{\mathrm{N}}}} . \, (\gamma + 1)\,\rho^{-\gamma-1}\, d\,\rho - \frac{d\,\rho}{2}.$$

Hence

$$\int_0^{\mathrm{N}} \rho\, d n = -\frac{\mathrm{N} + \frac{1}{2}\log_\epsilon \rho_{\mathrm{N}}}{1 - \frac{1}{\rho_{\mathrm{N}} x_{\mathrm{N}}}} \cdot \frac{\gamma + 1}{\gamma} \cdot \rho^{-\gamma}\Big]_0^{\mathrm{N}} - \tfrac{1}{2}\rho\Big]_0^{\mathrm{N}}$$

$$= \frac{\mathrm{N} + \frac{1}{2}\log_\epsilon \rho_{\mathrm{N}}}{1 - \frac{1}{\rho_{\mathrm{N}} x_{\mathrm{N}}}} \cdot \frac{\gamma + 1}{\gamma} \cdot \{1 - \rho_{\mathrm{N}}^{-\gamma}\} - \tfrac{1}{2}\rho_{\mathrm{N}} + \tfrac{1}{2}\rho_0.$$

Since $\rho^\gamma = x$, we get finally

$$\Sigma_1^{\mathrm{N}} \rho = \int_0^{\mathrm{N}} \rho\, d n + \tfrac{1}{2}\rho_{\mathrm{N}} - \tfrac{1}{2}\rho_0 = \frac{\mathrm{N} + \frac{1}{2}\log_\epsilon \rho_{\mathrm{N}}}{1 - \frac{1}{\rho_{\mathrm{N}} x_{\mathrm{N}}}} \cdot \frac{\gamma + 1}{\gamma}\left(1 - \frac{1}{x_{\mathrm{N}}}\right).$$

Values of

$$\frac{\gamma + 1}{\gamma} \frac{1 - \frac{1}{x}}{1 - \frac{1}{\rho x}}$$

for different values of x are given below, and this expression we may conveniently denote by b. Up to values of $x = 3$, the value of b is, with such ranges of turbine efficiency as are usual in practice, nearly independent of variations in the efficiency η of the turbines.

Thus comparative values of b for values of γ, equal respectively to 1.108 and 1.0675, are as follow:—

Pressure ratio, x, =	1.25	1.50	2.0	2.5
b (γ = 1.108)	1.1000	1.1796	1.2986	1.3836
b (γ = 1.0675	1.1039	1.1868	1.3108	1.3993

From Fig. 57 it will be seen that $\gamma = 1.108$ corresponds with steam, initially dry, to an efficiency of about 77 per cent., and $\gamma = 1.065$ to one of about 29 per cent. The difference between the corresponding values of b is, it will be seen, very small, even for pressure ratios as high as 2.5, whilst in practice this pressure ratio is commonly not much more than $\sqrt{2}$.

As proved on page 118, *ante*, the impulse on the blading of a group having no tip leakage is equal to

$$\frac{w}{g} \cdot \left[\frac{w}{\Omega} \cdot \cos a\ \Sigma_1^{\mathrm{N}} \mathrm{V} - \frac{\mathrm{N}s}{2}\right]$$

and $\Sigma_1^{\mathrm{N}} \mathrm{V} = \mathrm{V}_0\, \Sigma_1^{\mathrm{N}} \rho$, which, from the foregoing, is equal to $\mathrm{V}_0 (\mathrm{N} + \frac{1}{2}\log_\epsilon \rho)\, b \cos a$.

Hence the impulse may be written in the form

$$\frac{w}{g}\left[\frac{w \mathrm{V}_0}{\Omega} \cdot (\mathrm{N} + \tfrac{1}{2}\log_\epsilon \rho)\, b \cos a - \frac{\mathrm{N}s}{2}\right].$$

But $\frac{w\,V_0}{\Omega} = v_0$, the velocity at entrance, so that finally the impulse on the group is

$$\frac{w}{g}\left[v_0 (N + \tfrac{1}{2}\log_e \rho)\, b \cos\alpha - \frac{N\,s}{2}\right].$$

The work done per second is equal to this impulse multiplied by s, the blade speed. Whence W, the work done per pound of steam passed per second, is given by the relation

$$W = \frac{s}{g}\left\{v_0 (N + \tfrac{1}{2}\log_e \rho)\, b \cos\alpha - \frac{N\,s}{2}\right\}$$

It now remains to find an expression for the value of v_0, the velocity of the steam at entrance.

The energy tending to produce flow through any row of blades of the group is equal to the energy developed by the expansion of the steam as it flows through a row, plus the kinetic energy carried over from the preceding row.

Let q_n denote the expansion energy at any row n expressed in foot-pounds per pound of steam passed, and

$$\frac{r^2_{n-1}}{2\,g}$$

the kinetic energy "carried in" to the same row.

The kinetic energy of the steam as it leaves the row is equal to

$$\frac{v^2_n}{2\,g}$$

and we have then the relation

$$\frac{v^2_n}{2g} = m\,q_n + M\frac{r^2_{n-1}}{2\,g},$$

where M and m are coefficients to be determined from experiment.

Then if v_{n-1} be the velocity of outflow from row $(n-1)$, and s be the blade speed, we have from Fig. 50, page 79,

$$r^2_{n-1} = v^2_{n-1} + s^2 - 2\,s\,v_{n-1}\cos\alpha.$$

We thus get

$$\frac{v^2_n}{2\,g} = m\,q_n + \frac{M}{2\,g}\cdot\left[v^2_{n-} + s^2 - 2\,s\,v_{n-1}\cos\alpha\right].$$

Putting

$$v_n = \rho_n\,v_0$$

we get

$$\frac{v^2_n}{2\,g} = \frac{v^2_0\,\rho^2_n}{2\,g} = m\,q_n + \frac{M}{2\,g}\cdot\left[v^2_0\,\rho^2_{n-1} + s^2 - 2\,s\,v_0\,\rho_{n-1}\cos\alpha\right].$$

Now, since experiment has shown that former ideas as to

hydraulic shock are erroneous, we may take M as practically constant for every stage.

It is convenient to assume that there is a "carry-in" of energy even in the case of Stage No. 1, and to put this as equal to

$$\frac{M}{2g}(v^2_0 + s^2 - 2 s v_0 \cos \alpha).$$

The error involved at the most is very small, and the assumption makes the mathematics more symmetrical.

Adding up the values of $\frac{v^2_0 \rho^2_n}{2g}$ for every row, we get

$$\frac{v^2_0}{2g} \cdot \Sigma_1^N \rho^2_n = m \Sigma_1^N q_n + \frac{M}{2g} \cdot \left[v^2_0 \Sigma_0^{N-1} \rho^2_n + N s^2 - 2 s v_0 \cos \alpha \Sigma_0^{N-1} \rho_n\right].$$

On page 123, *ante*, it was shown that the

$$\Sigma_1^N \rho = \int_0^N \rho\, d n + \tfrac{1}{2}(\rho_N - 1),$$

and, similarly, by reference to Fig. 58 or Fig. 60, it will be obvious that

$$\Sigma_1^N \rho^2 = \int_0^N \rho^2 d n + \tfrac{1}{2}(\rho^2_N - \rho^2_0);$$

and ρ_0 is, of course, equal to 1.

Again, the figures in question show that

$$\Sigma_0^{N-1} \rho^2 \text{ is equal to } \int_0^N \rho^2 d n - \tfrac{1}{2}(\rho^2_N - 1),$$

and, similarly,

$$\Sigma_0^{N-1} \rho = \int_0^N \rho\, d n - \tfrac{1}{2}(\rho_N - 1).$$

The value of $\int_0^N \rho\, d n$ has already been found, and is equal to

$$b\,(N + \tfrac{1}{2} \log_\epsilon \rho_N) - \frac{\rho_N - 1}{2}.$$

Hence

$$\Sigma_0^{N-1} \rho = b\,(N + \tfrac{1}{2} \log_\epsilon \rho) - (\rho_N - 1).$$

To obtain the value of $\int_0^N \rho^2 d n$ we get from page 122, *ante*,

$$d n + \frac{d \rho}{2 \rho} = \frac{N + \tfrac{1}{2} \log_\epsilon \rho_N \,.}{1 - \frac{1}{\rho_N x_N}} (\gamma + 1) \,.\, \rho^{-\gamma - 2} d \rho.$$

Whence

$$\rho^2 d n + \tfrac{1}{2} \rho\, d \rho = \frac{N + \tfrac{1}{2} \log_\epsilon \rho_N \,.}{1 - \frac{1}{\rho_N x_N}} (\gamma + 1) \,.\, \rho^{-\gamma} d \rho.$$

Therefore

$$\int_0^x \rho^2\,dn = \frac{N + \frac{1}{2}\log_\epsilon \rho_x}{1 - \frac{1}{\rho_x x_x}} \cdot \frac{\gamma + 1}{1 - \gamma} \left.\rho^{1-\gamma}\right]_0^x - \left.\tfrac{1}{4}\rho^2\right]_0^x$$

$$= \frac{N + \frac{1}{2}\log_\epsilon \rho_x}{1 - \frac{1}{\rho_x x_x}} \cdot \frac{\gamma + 1}{\gamma - 1} \cdot \left(1 - \frac{\rho_x}{x_x}\right) - \frac{1}{4}\cdot\left(\rho^2_x - 1\right).$$

Substituting this value in our equation for $\frac{v^2_0}{2g}$ we get

$$\frac{v^2_0}{2g}\left\{(N + \tfrac{1}{2}\log_\epsilon \rho)\frac{\gamma+1}{\gamma-1} \cdot \frac{1 - \frac{\rho_x}{x_x}}{1 - \frac{1}{\rho_x x_x}} + \tfrac{1}{4}(\rho^2_x - 1)\right\}$$

$$= m\,\Sigma_0^x q_x + \frac{M\,v^2_0}{2g} \cdot \left\{\begin{array}{c} (N + \frac{1}{2}\log_\epsilon \rho)\dfrac{\gamma+1}{\gamma-1} \cdot \dfrac{1 - \frac{\rho_x}{x_x}}{1 - \frac{1}{\rho_x x_x}} - \frac{3}{4}(\rho^2 - 1) + N\,\delta^2 \\ - 2\,\delta\, v_0 \cos\alpha\,\{b\,(N + \frac{1}{2}\log_\epsilon \rho) - \rho_x + \rho_0\} \end{array}\right\} \quad (35)$$

Now $\Sigma_0^x q_n$ is the total work done in expansion,

$$i.e., \ \Sigma\, q_x = \frac{\gamma}{\gamma - 1} \cdot P_0 V_0\left(1 - \frac{\rho_x}{x_x}\right).$$

We have thus obtained a rational expression connecting together the steam-speed at entrance to a group, with ρ the total ratio of expansion in the group, s the blade speed, and N the number of rows in the group.

It was stated above that the value of this velocity was very little affected by such variations in the value of γ as occur in practice, and this is well shown by Table X., in which comparison has been made between the value of v_0^2 when $\gamma = 1.0675$ and its value when $\gamma = 1.108$. The former value of γ corresponds, as shown by Fig. 57, to a turbine efficiency of somewhere under 30 per cent., and the latter to an efficiency of about 77 per cent. As will be seen, the two sets of figures are in very close agreement, the extreme range in the value of v not exceeding about 1 per cent. even for large values of x, and being much less for such values of this ratio as are usual in practice.

With the formulas above established it is possible to determine separately the output and efficiency of each group of a reaction turbine in practically the same way as the corresponding figures are obtained in designing compartment-compounded impulse turbines.

TABLE X.—VALUES OF v_0 FOR DIFFERENT CONDITIONS OF EXPANSION, BLADE SPEED AND NUMBER OF ROWS.

—	$\delta = \frac{s}{v_0}$.			0.4	0.6	0.8
$x = 1.25$	N = 6	$\gamma = 1.0675$	$v_0^2 = p_0 V_0 \times$	312.70	282.92	267.09
		$\gamma = 1.108$		313.83	283.93	268.11
	N = 30	$\gamma = 1.0675$	$v_0^2 = p_0 V_0 \times$	65.555	58.911	55.020
		$\gamma = 1.108$		65.580	58.890	54.985
$x = 1.50$	N = 6	$\gamma = 1.0675$	$v_0^2 = p_0 V_0 \times$	469.50	427.68	403.90
		$\gamma = 1.108$		474.68	429.40	408.29
	N = 30	$\gamma = 1.0675$	$v_0^2 = p_0 V_0 \times$	102.88	92.510	86.172
		$\gamma = 1.108$		103.24	92.851	86.546
$x = 2.0$	N = 6	$\gamma = 1.0675$	$v_0^2 = p_0 V_0 \times$	600.00	553.74	525.30
		$\gamma = 1.108$		611.26	563.57	534.46
	N = 30	$\gamma = 1.0675$	$v_0^2 = p_0 V_0 \times$	141.68	127.98	119.08
		$\gamma = 1.108$		142.39	128.60	119.69
$x = 2.5$	N = 6	$\gamma = 1.0675$	$v_0^2 = p_0 V_0 \times$	633.14	591.01	563.92
		$\gamma = 1.108$		648.84	604.80	576.81
	N = 30	$\gamma = 1.0675$	$v_0^2 = p_0 V_0 \times$	160.02	145.25	135.30
		$\gamma = 1.108$		160.94	146.06	136.11

In making such a calculation, however, it will be found in every way preferable to start at the exhaust end and work backwards. The fact is that large variations in the vacuum hardly affect at all the inflow, into the first group of a reaction turbine of the usual type, under a given initial pressure. Conversely, very small differences in the weight passed, under a given pressure, through the first row of blades, correspond to very large differences in the final discharge pressure from the last group.

Starting from the high-pressure end and computing the turbine group by group down to the exhaust is almost analogous to working with a divergent series in algebra, whilst if the computation is started at the exhaust end, the operation may be likened to the use of a highly convergent series. In making such a calculation the curves plotted in Figs. 53 to 56, page 98, may be used. These are all based on the condition of the steam on discharge from a group in place of its condition at entrance. They have been derived directly from equations (34) and (35) by taking γ as

1.108. For those who wish to plot the curves to a larger scale, the values of the coefficients are given in Tables XI., XII., and XIII.,* on pages 129 and 130.

Having obtained the value of v_N, as in the examples worked out in Chapter XII., *ante*, the work done by the group is obtained from the formula

$$\frac{h - c}{h + \sigma c} \cdot \frac{s}{32.2} \cdot \left\{ v_N \,.\, \text{B} \,.\, (\text{N} + 1.05 \log x) \cos \alpha - \frac{\text{N}\, s}{2} \right\},$$

where B is taken from the curve plotted in Fig. 56.

This value of B is obtained from the same formula as was used to find b, as tabulated on page 123, by substituting the conditions at discharge for those at entrance to the group.

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* In plotting such curves to a large scale the use of formulæ of interpolation is often very convenient, as it is not always easy to get a curve or a spline to fit the tabulated values when widely separated. The most useful of these formulæ is as follows:—If y_0, y_1, y_2, and y_3 be four consecutive values of y, for four equidistant values of x, which may be called x_0, x_1, x_2, and x_3, then the value of y, corresponding to $x_{1.5}$—*i.e.*, to a value of x, midway between x_1 and x_2—is

$$y_{1.5} = \tfrac{9}{16}(y_1 + y_2) - \tfrac{1}{16}(y_0 + y_3).$$

The calculation is most conveniently made as follows:—Add y_1 and y_2, and take the mean. Call this mean $\bar{y}$. Also add y_0 and y_3, and take the mean of these, calling this second mean y^1; then the required value of $y_{1.5}$ is $\bar{y} + \frac{1}{8}(\bar{y} - y^1)$. This formula was published as new in the *Comptes Rendus* in 1911, but had previously been given in ENGINEERING, vol. lxiii., page 398. It is easily deduced from Lagrange's formula of interpolation. As an example, suppose a value of ϕ (Table XI.) is needed for $x = 2.125$ and $\delta_x = 0.2$. We have from the table, $y_0 = 1.1559$, $y_1 = 1.6390$, $y_2 = 2.1689$, $y_3 = 2.7460$.

Then

$$\bar{y} = \frac{1.6390 + 2.1689}{2} = 1.9040$$

$$y^1 = \frac{1.1559 + 2.7460}{2} = 1.9510$$

and

$$\bar{y} - y^1 = -0.0470.$$

Hence the required value of y is

$$y_{1.5} = \bar{y} + \frac{\bar{y} - y^1}{8} = 1.9040 - 0.0059 = 1.8981.$$

If the values of x are not equidistant, then we may calculate $x_{1.5}$ corresponding to the above value of $y_{1.5}$ by an exactly similar formula—viz., $x_{1.5} = \bar{x} + \frac{\bar{x} - x^1}{8}$.

Obviously the formula given above cannot be used to calculate, in the case of Table XI., say, values of y corresponding to $x = 1.375$ or $x = 2.375$, but for these we may use the formula

$$y_{0.5} = \tfrac{5}{16}\left[y_0 + 3\,y_1 - y_2 + \frac{y_3}{5}\right].$$

TABLE XI.—STANDARD BLADES. VALUES OF ϕ AND μ FOR PLOTTING THE CURVES IN FIG. 53.

Values of $x = \frac{p_0}{p_x}$		1.25	1.5	1.75	2.0	2.25	2.5
$\delta_x = 0.2$	ϕ	0.3344	0.7206	1.1559	1.6390	2.1689	2.7460
	μ	0.4167	0.7526	1.0491	1.3211	1.5836	1.8250
$\delta_x = 0.3$	ϕ	0.3016	0.6484	1.0386	1.4722	1.9486	2.4683
	μ	0.3619	0.6498	0.9006	1.1282	1.3468	1.5478
$\delta_x = 0.4$	ϕ	0.2789	0.6000	0.9625	1.3670	1.8144	2.3071
	μ	0.3216	0.5743	0.7938	0.9931	1.1851	1.3600
$\delta_x = 0.5$	ϕ	0.2630	0.5674	0.9137	1.3040	1.7404	2.2262
	μ	0.2909	0.5183	0.7361	0.8955	0.9479	1.2290
$\delta_x = 0.6$	ϕ	0.2522	0.5468	0.8859	1.2734	1.7130	2.2112
	μ	0.2672	0.4757	0.6573	0.8238	0.9868	1.1378

TABLE XII.—SEMI-WING BLADES. VALUES OF ϕ AND μ FOR PLOTTING CURVES IN FIG. 54.

Values of $x = \frac{p_0}{p_x}$		1.25	1.5	1.75	2.0	2.25	2.5
$\delta_x = 0.2$	ϕ	0.3437	0.7419	1.2042	1.7244	2.2972	2.9004
	μ	0.4326	0.7847	1.1059	1.4073	1.7000	1.9568
$\delta_x = 0.3$	ϕ	0.3097	0.6747	1.0987	1.5761	2.1016	2.6598
	μ	0.3772	0.6872	0.9695	1.2330	1.4864	1.7078
$\delta_x = 0.4$	ϕ	0.2921	0.6318	1.0307	1.4885	1.9990	2.5345
	μ	0.3440	0.6179	0.8718	1.1132	1.3494	1.5451
$\delta_x = 0.5$	ϕ	0.2777	0.6014	0.9917	1.4446	1.9558	2.4962
	μ	0.3159	0.5671	0.8032	1.0306	1.2555	1.4408
$\delta_x = 0.6$	ϕ	0.2686	0.5851	0.9757	1.4382	1.9702	2.5411
	μ	0.2945	0.5296	0.9043	0.9746	1.1968	1.3842

TABLE XIII.—WING BLADES. VALUES OF ϕ AND μ FOR PLOTTING THE CURVES IN FIG. 55

Values of $x = \frac{p_0}{p_x}$		1.25	1.5	1.75	2.0	2.25	2.5
$\delta_x = 0.2$	ϕ	0.3555	0.7693	1.2388	1.7632	2.3419	2.9754
	μ	0.4494	0.8168	1.1523	1.4700	1.7838	2.0718
$\delta_x = 0.3$	ϕ	0.3279	0.7089	1.1400	1.6260	2.1730	2.7940
	μ	0.4022	0.7279	1.0156	1.2856	1.5499	1.8123
$\delta_x = 0.4$	ϕ	0.3096	0.6706	1.0831	1.5489	2.0696	2.6496
	μ	0.3680	0.6650	0.9285	1.1729	1.4125	1.6385
$\delta_x = 0.5$	ϕ	0.2977	0.6481	1.0530	1.5160	2.0419	2.6379
	μ	0.3426	0.6200	0.8676	1.0995	1.3298	1.5512
$\delta_x = 0.6$	ϕ	0.2912	0.6389	1.0466	1.5228	2.0748	2.7156
	μ	0.3243	0.5888	0.8277	1.0555	1.2865	1.5152

CHAPTER XV.

RADIAL-FLOW REACTION TURBINES.

IN the case of a radial-flow turbine, such as the Ljungström machine, of which a detailed description is given elsewhere in this volume, the passage of the working fluid through the blading is in part due to centrifugal forces, and thus the simple diagram of velocities which suffices for the analysis of an axial-flow machine requires to be replaced by another method of treatment.

Provisional estimates of the output and capacity of a radial-flow machine may, however, be made in the same way as with axial-flow machines, by means of the formula given in Chapter VII., viz :—

$$N \cdot \left[\frac{d}{10}\right]^2 \cdot \left[\frac{\text{R.P.M.}}{100}\right]^2 = \lambda.$$

Here, however, d is to be taken as the outer diameter in inches of the smallest ring of blades, R.P.M. denotes the number of revolutions of one disc taken relatively to that of the other, and N the number of rows which would be required on one disc for a given value of the coefficient λ if all these rings had the same diameter d. If λ be assumed as 180,000, which, from the curve, Fig. 43, page 50, will correspond to a brake efficiency ratio of somewhere about 74 per cent., the value of N, taking d as 10 in. and R.P.M. as 6000, is 50. In that case, the velocity of inflow through the first row of blades is, as a rough approximation, given by the semi-empirical relation already used in Chapter VII., viz.:—

$$v = \frac{2675}{\sqrt{N}} = 380 \text{ ft. per second nearly.}$$

From this the steam-way needed at the first row can be determined in the usual way.

Actually, of course, the different rings of blading are necessarily of different diameters, so that the numbers of rows really required

will not be N, but a much smaller number, N_1 say, which is given approximately by the relation

$$N_1 \cdot \frac{D^2 + d^2 + (d + D)^2}{6} = N d^2 - \tfrac{1}{2}(D^2 - d^2),$$

where D denotes the outer diameter of the outermost ring. If this be 30 in., for example, we get

$$N_1 \cdot \frac{900 + 100 + 1600}{6} = 5000 - \tfrac{1}{2}(900 - 100),$$

whence

$$\mathbf{N}_1 = \frac{6 \times 4600}{2600} = 11 \text{ nearly},$$

so that eleven rings of blades per disc would suffice to give the high coefficient of 180,000.

Having thus obtained the general dimensions of a radial-flow turbine, its performance can be analysed in detail by making use of the relation between torque and changes in angular momentum, which are given in treatises on theoretical mechanics.

Suppose, accordingly, that a pound of fluid is delivered per second from a ring of fixed blades with a velocity of outflow equal to v_1. Then this velocity can be resolved into a tangential component $t_1 = v_1 \cos \alpha$, where α is the blade angle at discharge, and a radial component $a_1 = v_1 \sin \alpha$. The tangential momentum of the pound of fluid as delivered will be equal to $\frac{1}{g} t_1$, and if r_1 be the radius of the discharge edges of the blades, the moment of this momentum about the centre of the shaft will be

$$\frac{1}{g} t_1 r_1.$$

Similarly, when the fluid is discharged through the next row of blades, its absolute tangential momentum will be $\frac{1}{g} t_2 r_2$, where t_2 denotes its new tangential velocity, which, as this row of blades is moving, will be given by the relation

$$t_2 = v_2 \cos \alpha - r_2 \omega,$$

where ω denotes the angular velocity of the disc.

Generally t_1 and t_2 will be directed in opposite directions, so that the change effected per second in the moment of momentum of the pound of fluid will be

$$\frac{1}{g} \cdot (t_1 r_1 + t_2 r_2).$$

Now by a well-known principle in theoretical mechanics this is numerically equal to the torque T, which has acted on the pound of fluid during its passage from r_1 to r_2, and since action and reaction are equal and opposite, this represents also the torque acting on the moving row of blades. The work done by this row per pound of fluid passed per second is therefore

$$W = T\omega = \frac{\omega}{g}(t_1 r_1 + t_2 r_2) \text{ ft.-lb.}$$

The equations governing the flow of the steam through the blading are perhaps most easily established by equating the total energy of the steam before it enters a row of blades to the work it does in the next row plus its total energy after discharge.

Thus, let H_1 be the total heat, measured in foot-pounds, contained in 1 lb. of steam as it issues from Row No. 1, which may, without loss of generality, be considered to be a fixed row. If its velocity on issue is v_1, its total energy is $H_1 + \frac{v_1^2}{2g}$. If H_2 be its total heat on discharge from Row No. 2, and l its then absolute velocity, its total energy on discharge will be $H_2 + \frac{l_2^2}{2g}$. At the same time as No. 2 is a moving row, work has been done by it equal to

$$\frac{\omega}{g}\left(r_1 t_1 + r_2 t_2\right) = \frac{\omega}{g}\left(r_1 v_1 \cos\alpha + r_2 v_2 \cos\alpha - r_2^2 \omega\right).$$

Hence

$$H_1 + \frac{v_1^2}{2g} = H_2 + \frac{l_2^2}{2g} + \left(r_1 v_1 \cos\alpha + r_2 v_2 \cos\alpha - r_2^2\omega\right)\frac{\omega}{g}.$$

Since

$$l_2^2 = v_2^2 + \omega^2 r_2^2 - 2 r_2 v_2 . \omega . \cos\alpha.$$

This reduces to

$$H_1 + \frac{v_1^2}{2g} = H_2 + \frac{v_2^2}{2g} - \frac{r_2^2\omega^2}{2g} + \frac{\omega}{g} . r_1 v_1 \cos\alpha.$$

This equation holds whether there are frictional losses or not in Row No. 2, since any such friction goes to increase the value of H_2. Let H_2^1 be the value which H_2 would have if there were no frictional losses. Then we get

$$H_1 + \frac{v_1^2}{2g} = H_2^1 + \frac{v_2^2}{2g} - \frac{r_2^2\omega^2}{2g} + \frac{\omega}{g} r_1 v_1 \cos\alpha + \text{friction}.$$

The frictional losses are equal to

$$(1 - m)(H_1 - H_2^1) + (1 - M) \times \text{the "carry-over"}$$

from Row No. 1, where m and M are coefficients.

This "carry-over" from the first ring taken relatively to the second is equal to

$$\frac{1}{2g}(v_1^2 + r_1^2\omega^2 - 2 v_1 r_1 . \omega \cos \alpha).$$

Substituting this value and rearranging the terms of our equation we get finally

$$m(H_1 - H_2^1) = \frac{v_2^2}{2g} + \frac{r_1^2\omega^2}{2g} - \frac{r_2^2\omega^2}{2g} - \frac{M}{2g}(v_1^2 + r_1^2\omega^2 - 2 v_1 r_1 \omega \cos \alpha).$$

If we next take a fixed row, say Row No. 3, the total energy of the steam on discharge from Row No. 2 is as stated above.

$$H_2 + \frac{v_2^2}{2g} + \frac{\omega^2 r_2^2}{2g} - \frac{2 r_2 v_2 \omega . \cos a}{2g},$$

whilst on discharge from Row No. 3 its total energy will be $H_3 + \frac{v_3^2}{2g}$. Since the row is fixed, no energy is abstracted in the form of work on the shaft, and we thus have

$$H_2 + \frac{v_2^2}{2g} + \frac{\omega^2 r_2^2}{2g} - \frac{2 r_2 v_2 \omega \cos a}{2g} = H_3 + \frac{v_3^2}{2g}.$$

As before, let H_3^1 denote the value which H_3 would have if there were no frictional loss. Then

$$H_2 + \left(\frac{v^2}{2g} + \frac{\omega^2 r_2^2}{2g} - \frac{2 r_2 v_2 \omega \cos a}{2g}\right)$$

$$= H_3^1 + \frac{v_3^2}{2g} + (1 - m)(H_2 - H_3^1) + (1 - M) \times \text{"carry-over"}$$

from Row No. 2. This carry-over from Row No. 2 is

$$\frac{v_2^2 + \omega^2 r_2^2 - 2 r_2 v_2 \omega \cos a}{2g}.$$

Hence, on substituting this value and rearranging, we get

$$m(H_2 - H_3^1) = \frac{v_3^2}{2g} - \frac{M}{2g}(v_2^2 + r_2^2\omega^2 - 2 r_2 v_2 \omega \cos \alpha).$$

Forming similar equations for each row, we get

$$\begin{aligned}
2gm(H_0 - H_1^1) &= v_1^2 \\
2gm(H_1 - H_2^1) &= v_2^2 + r_1^2\omega^2 - r_2^2\omega^2 - M(v_1^2 + r_1^2\omega^2 - 2 v_1 r_1 \omega \cos \alpha) \\
2gm(H_2 - H_3^1) &= v_3^2 \qquad\qquad - M(v_2 + r_2^2\omega^2 - 2 v_2 r_2 \omega \cos \alpha) \\
&\cdots\cdots\cdots \\
2gm(H_{n-2} - H^1_{n-1}) &= v^2_{n-1} - M(v^2_{n-2} + r^2_{n-2}\omega^2 - 2 v_{n-2} . r_{n-2}\omega \cos \alpha) \\
2gm(H_{n-1} - H^1_n) &= v^2_n + r^2_{n-1} . \omega^2 - r^2_n \omega^2 - M(v^2_{n-1} + r^2_{n-1}\omega^2 - \\
&\qquad 2 v_{n-1} . r_{n-1}\omega \cos \alpha).
\end{aligned}$$

The sum of all the quantities on the left is equal to the total heat

energy theoretically available in the turbine multiplied by the reheat factor. If this total is equal to U heat units, we have, finally,

$$2g\,J\,.\,m\,.\,U = \sum_{1}^{N} v^2 + \omega^2\,.\sum_{n=0}^{n=\frac{N-2}{2}} r^2_{2n+1} - \omega^2 \sum_{n=1}^{n=\frac{N}{2}} r^2_{2n} - M \sum_{n=1}^{n=N-1} (v^2_n + r^2_n\,\omega^2 - 2r_n\,v_n\,\omega\,.\cos\alpha)$$

In applying this equation two cases naturally arise. Thus the blade may be of constant length, or else have its length adjusted so that the ratio of blade speed to steam speed is constant throughout. Where feasible this latter arrangement is obviously desirable, and it is that which will be considered here. In that case we may put $v = C\,r$, and we then get

$$2g\,J.m.U = C^2 \sum_{1}^{N} r^2 + \omega^2 \sum_{n=0}^{n=\frac{N-2}{2}} r^2_{2n+1} - \omega^2 \sum_{n=1}^{n=\frac{N}{2}} r^2_{2n} - M\,(C^2 + \omega^2 - 2\,C\,\omega) \sum_{n=1}^{n=N-1} r^2_n \quad . \quad . \quad (36)$$

As an example, assume the steam to be supplied dry and at 165 lb. absolute, and exhausted at 28 in. vacuum. Then the heat theoretically available is about 321 B.Th.U. If the reheat factor be 1.057, we get for U, the total heat, which becomes available during the passage of the steam through the turbine, U = 1.057 × 321 = 339 B.Th.U. per lb.

We shall, as before, take m = 0.89 and M = 0.50.

It is convenient to take the pitch of the rows in even dimensions, so that we will take the radius of the outer row as 15½ in. instead of the 15 in. originally assumed. This will slightly increase the coefficient λ and, consequently, the efficiency; though with such a small change as this the alteration will not be great.

If the pitch be taken as uniform at ½ in., the various summations in equation (36) can be easily effected by means of the calculus of finite differences; but with such a small number of rows, this is hardly worth while, and it is really simpler to tabulate the quantities involved as below, and add them up direct, as in Table XIV., on page 136.

Since the radii have been taken in inches instead of in feet, the totals arrived at should be divided by 144, but it is simpler to multiply the left-hand side of equation (36) by 144. We thus get

$$144 \times 64.4 \times 778 \times 0.89 \times 339$$
$$= 2532.75\ C^2 - \omega^2\,[1322.75 - 1210] - 0.50 \times 2292.50\,[C^2 + \omega^2 - C\,\omega\cos\alpha]$$
$$\therefore 144 \times 64.4 \times 778 \times 0.89 \times 339$$
$$= 2532.75\,C^2 - \omega^2\,[1322.75 - 1210] - 0.50 \times 2292.50\,[C^2 + \omega^2 - C\,\omega\cos\alpha].$$

TABLE XIV.—CALCULATION OF RADIAL-FLOW TURBINE.

No. of Row.	Radius of Ring.	[Radius of Ring]^2.	[Odd Radii]^2.	[Even Radii]^2.
	in.	sq. in.	sq. in.	sq. in.
1	5.0	25.00	25.00	—
2	5.5	30.25	...	30.25
3	6.0	36.00	36.00	—
4	6.5	42.25	...	42.25
5	7.0	49.00	49.00	—
6	7.5	56.25	...	56.25
7	8.0	64.00	64.00	—
8	8.5	72.25	...	72.25
9	9.0	81.00	81.00	—
10	9.5	90.25	...	90.25
11	10.0	100.00	100.00	—
12	10.5	110.25	...	110.25
13	11.0	121.00	121.00	—
14	11.5	132.25	...	132.25
15	12.0	144.00	144.00	—
16	12 5	156.25	...	156.25
17	13.0	169.00	169.00	—
18	13.5	182.25	...	182.25
19	14.0	196.00	196.00	—
20	14.5	210.25	...	210.25
21	15.0	225.00	225.00	—
22	15.5	240.25	...	240.25
Total ...	...	2532.75	1210.00	1322.75

If the relative motion of one disc to the other is 6000 revolutions per minute, then $\omega = \dfrac{2\,\pi \times 6000}{60} = 628.3$. Hence, if a is taken as 20 deg., the above equation reduces to

$$1{,}570{,}000 = C^2 - 294{,}260 + 1038.9\ C.$$

Whence

$$C = 940.$$

Hence the speed of inflow through the first row of blades is $940 \times \frac{5}{12} = 392$ ft. per second, as against 380, given by the approximate rule.

The indicated work done per pound of steam passed is equal to

$$\frac{\omega}{g}\left[\sum_{1}^{N} r\,v\cos\alpha - \omega\,.\sum_{n=1}^{n=\frac{N}{2}} r_{2n}^{2}\right] \quad . \quad . \quad . \quad . \quad . \quad (37)$$

Putting $v = r\,C$.

This becomes

$$\frac{628.3}{144 \times 32.2}\left[940 \times 0.9397 \times 2532.75 - 628.3 \times 1322.75\right] \text{ ft.-lb.,}$$

or 244.9 B.Th.U.

Hence, if there were no leakage losses, the indicated efficiency ratio would be

$$\frac{244.9}{321} = 0.763,$$

which agrees very well with the brake efficiency ratio deduced from the curve, Fig. 43, page 50.

No allowance has, however, been made for leakage losses; but, as against this, it will be remembered that the values of m and M were purposely taken a little small, and further that both coefficients probably increase with increases in the steam speed. Hence the detailed calculation and the curve may be considered to be in very satisfactory agreement.

From equation (37) it is evident that the efficiency will be a maximum when

$$C\cos\alpha\,.\sum_{1}^{N} r^{2} = 2\,\omega\sum_{n=1}^{n=\frac{N}{2}} r^{2}_{n},$$

which in this case gives $\omega = 846$ radians per second. The corresponding best ratio of blade speed to steam speed is about 0.9. The coefficient λ would then be about 340,000, and the corresponding indicated efficiency ratio would be brought up to about 82 per cent. The use of superheat would, as is well known, increase this.

It was assumed in the foregoing that the blade lengths were adjusted so that the blade speed was a constant fraction of the steam speed. This adjustment is best made by means of the approximate equation to the expansion of the steam, viz.:—

$$p\,V^{\gamma} = p_0\,V_0^{\gamma}\,.$$

Initially we have $p_0 = 165$, and $V_0 = 2.757$ cub. ft. per lb. As originally supplied, the steam had a heat content of 1192.9 B.Th.U. per lb. Of this total energy, 244.9 B.Th.U. were removed from

the steam as useful work done by the blades, so that on discharge the heat content of the steam was 1192.9 − 244.9 = 948 B.Th.U. A reference to the Mollier diagram, Fig. 7, facing page 8, shows that the steam was therefore about 0.842 dry, so that $V_N = 340 \times 0.842 = 286$ cub. ft. per lb. The exhaust pressure being 2 in. absolute, we have $p_N = 0.98$ lb. Hence, to determine γ, we have

$$0.98 \times 286^{\gamma} = 165 \times 2.757^{\gamma},$$

so that

$$\gamma = \frac{\log 165 - \log . 0.98}{\log 286 - \log 2.757} = 1.1041.$$

The total energy developed by the expansion of the steam is then equal to

$$\frac{\gamma}{\gamma - 1} \cdot (P_0 V_0 - P_N V_N) = \frac{1.1041}{0.1041} P_0 V_0 (1 - A_N),$$

where P denotes pressures in pounds per square foot.

Now from the fact that the ratio of the blade speed to the steam speed is kept constant it will be obvious that the total energy expended after passing through the nth row is equal to

$$U \times \frac{r_1^2 + r_2^2 + \&c. + r_n^2}{r_1^2 + r_2^2 + \&c. + r_N^2} = \frac{1.1041}{0.1041} P_0 V_0 (1 - A_n),$$

Whence

$$\frac{1 - A_n}{1 - A} = \frac{\sum_1^n r^2.}{\sum_1^N r^2}$$

Now

$$A_N = \frac{p_N V_N}{p_0 V_0} = \frac{280.28}{454.90} = 0.61614,$$

Therefore

$$A_n = 1 - \frac{0.38386 \sum_1^n r^2}{\sum_1^N r^2}.$$

Hence the value of A corresponding to Row No. 8 is

$$A_8 = 1 - \frac{0.38386 \times 375.00}{2532.75} = 0.94317.$$

Similarly,

$$A_{16} = 1 - \frac{0.38386}{2532.75} . 1310.00 = 0.80146.$$

The volume of the steam at any point is then given by the relation

$$V_n = V_0 \left[\frac{1}{A_n} \right]^{\frac{1}{\gamma - 1}} = V_0 \left[\frac{1}{A_n} \right]^{\frac{1}{0.1041}}$$

Hence

$$V_8 = 4.837, \text{ and } V_{16} = 23.11 \text{ cub. ft. per lb.}$$

The steam speeds at the corresponding radius are equal to $\frac{C \,.\, r}{12}$ if r be taken in inches, so that, since C was found to be 940, we get

$$v_8 = 665 \text{ ft. per second, } v_{16} = 979 \text{ ft. per second, and } v_{22} = 1173 \text{ ft. per second.}$$

The area available for flow at any ring is, of course, $2\,\pi\, r\, h \sin a$, where h denotes the blade length. The corresponding steam speed is Cr. Hence, if $\dot{w}$ be the weight passed per second, we must have at every row $2\,\pi\, r\, h \sin a \times \mathrm{C}\, r = w\,\mathrm{V}$. Hence h varies as $\frac{\mathrm{V}}{r^2}$.

If for convenience we assume an imaginary row corresponding to a radius $r_0 = 4.5$ in., then, if h_0 be the length of blade which would be required here, we have

$$h_8 = h_0 \,.\, \frac{r_0^2}{r_8^2} \,.\, \frac{V_8}{V_0} = h_0 \frac{20\ 25}{72.25} \,.\, \frac{4.837}{2.757} = 0.542\, h_0.$$

$$h_{16} = h_0 \,.\, \frac{20.25}{156.25} \,.\, \frac{23.11}{2.757} = 1.086\, h_0.$$

$$h_{22} = h_0 \,.\, \frac{20.25}{240.25} \,.\, \frac{286}{2.757} = 8.74\, h_0.$$

Intermediate values may be found with sufficient accuracy by drawing a smooth curve through h_0, h_8, h_{16}, and h_{22}.

In the foregoing it has been assumed that a "double-rotation" turbine can be replaced for the purpose of calculation by a "single-rotation" turbine running at double the speed. This is true everywhere save at the first and last rows of blades; but unless special provision is made, the loss at the last row by "carry-over" to the exhaust will be greater with the double-rotation type than with the equivalent single-rotation turbine.

CHAPTER XVI.

THERMODYNAMIC PRINCIPLES.

IN 1824 Sadi Carnot conceived the ideal heat engine of maximum efficiency of which the working cycle is fully described in most text books on ordinary steam or gas engines.

By using this ideal engine as a basis for discussion, it is easy to show that when a working medium receives heat at the absolute temperature T, the fraction of this heat which is available for conversion into the form of mechanical work is equal to $\frac{T - T_2}{T}$ where T_2 denotes the lowest working temperature measured on the absolute scale. Thus, when steam is produced from boiling water, the latent heat L is added at a constant temperature, so that the fraction of this latent heat which is available for conversion into work is $L\left(\frac{T - T_2}{T}\right)$

This may also be written in the form

$$L - T_2 \cdot \frac{L}{T}.$$

The quantity $\frac{L}{T}$ is known as the increase of entropy due to the heat added in the process of evaporation, or more shortly as the steam entropy, and may be denoted by ϕ_s.

In general, however, when heat is added to a working medium the temperature does not remain constant.

Thus, taking water at freezing point, this must be raised to the temperature T, before evaporation, under the pressure corresponding to this temperature, can proceed. At any given instant, during this process of warming up the water, let the temperature be T. If this temperature be now raised by the very small amount d T, the heat which must be added to effect this rise is equal to $\sigma \cdot d$ T where σ denotes the specific heat of the water at

the given temperature T. Of this total amount, only the fraction $\sigma \, . \, d\,T \left(1 - \frac{T_2}{T}\right)$ is available for conversion into mechanical work. Hence of the total heat added in heating up 1 lb. of water to the temperature of evaporation, the amount available for work production is

$$\int_{T_2}^{T_1} \sigma \, d\,T - T_2 \int_{T_2}^{T_1} \sigma \, . \, \frac{d\,T}{T} \quad . \quad . \quad . \quad . \quad . \quad (38)$$

Where T_1 denotes the temperature at which the steam is finally evaporated, and T_2, as before, the lowest working temperature.

Obviously

$$\int_{T_2}^{T_1} \sigma \, d\,T = h_1 - h_2$$

Where h_1 denotes the sensible heat of saturated steam at T_1, and h_2 the sensible heat of saturated steam at the temperature T_2.

The term $\int_{T_2}^{T_1} \sigma \, . \, \frac{d\,T}{T}$ is known as the increase of entropy due to heating 1 lb. of water from the temperature T_2 to the temperature T_1, and may be expressed as $\phi_{l_1} - \phi_{l_2}$ where ϕ_{l_1} denotes the increase in entropy due to heating up the water from freezing point to T_1, and ϕ_{l_2} the increase in entropy on heating 1 lb. of water from freezing point to T_2. More briefly ϕ_{l_1} is the liquid entropy at T_1 and ϕ_{l_2} the liquid entropy at T_2. Values of ϕ_l are tabulated for a number of different temperatures in Table XV., page 142, as well as the corresponding values of $\phi_s = \frac{L}{T}$. The term ϕ is known as the total entropy of the pound of steam, and is given by the relation

$$\phi = \phi_l + \phi_s.$$

It thus appears that with dry saturated steam supplied at a temperature T_1, the fraction of the total heat which is available for conversion into work is given by the relation

$$\begin{aligned} u &= h_1 - h_2 - T_2 (\phi_{l_1} - \phi_{l_2}) + L_1 - T_2 \, . \, \phi_{s_1} \\ &= h_1 - h_2 - T_2 (\phi_{l_1} - \phi_{l_2}) + \phi_{s_1} (T_1 - T_2) \quad . \quad . \quad . \quad (39) \end{aligned}$$

All of the quantities involved are given in a modern steam table from which Table XV. is an abstract.

Putting

$$H_1 = h_1 + L_1 \, ; \, H_2 = h_2 + L_2$$

so that H denotes the total heat at the temperature T; equation (39) can also, as shown later, be written in the simpler form

$$u = H_1 - H_2 + T_2 (\phi_1 - \phi_2) \quad . \quad . \quad . \quad . \quad (40)$$

TABLE XV.—THE PROPERTIES OF DRY SATURATED STEAM AT DIFFERENT ABSOLUTE TEMPERATURES.

T. Absolute Temperature Fahr.	P. Absolute Pressure in Pounds per Square Inch.	h. Sensible Heat. B.Th.U.	L. Latent Heat. B.Th.U.	H. Total Heat. B.Th.U.	ϕ_l, Liquid Entropy.	$\Delta \phi_l$, Increment per Deg.	ϕ_s, Steam Entropy.	$\Delta \phi_s$, Decrement per Deg.	ϕ, Total Entropy.	$\Delta \phi$, Decrement per Deg.	Specific Volume of Steam in Cubic Feet.
520	0.2627	28.89	1073.66	1102.55	0.0561	0.00188	2.0646	0.00524	2.1207	0.00366	1173.5
530	0.3731	38.76	1066.51	1105.27	0.0749	0.00185	2.0122	0.00505	2.0871	0.00320	839.7
540	0.5213	48.65	1059.37	1108.02	0.0934	0.00182	1.9617	0.00486	2.0551	0.00304	610.9
550	0.7169	58.56	1052.23	1110.79	0.1116	0.00179	1.9131	0.00469	2.0247	0.00290	452.0
560	0.9735	64.49	1045.09	1113.58	0.1295	0.00176	1.8662	0.00453	1.9957	0.00277	338.6
570	1.306	78.43	1037.95	1116.38	0.1471	0.00173	1.8209	0.00437	1.9680	0.00264	256.7
580	1.731	88.39	1030.80	1119.19	0.1644	0.00171	1.7772	0.00422	1.9416	0.00248	196.7
590	2.271	93.37	1023.66	1122.03	0.1815	0.00168	1.7353	0.00408	1.9168	0.00243	152.3
600	2.948	108.37	1016.52	1124.89	0.1983	0.00165	1.6942	0.00395	1.8925	0.00230	119.1
610	3.790	118.38	1009.38	1127.76	0.2148	0.00163	1.6547	0.00382	1.8695	0.00219	94.07
620	4.925	128.41	1002.24	1130.65	0.2311	0.00161	1.6165	0.00370	1.8476	0.00209	74.96
630	6.089	138.46	995.10	1133.56	0.2472	0.00158	1.5795	0.00358	1.8267	0.00200	60.25
640	7.622	148.52	987.95	1136.47	0.2630	0.00156	1.5437	0.00347	1.8067	0.00191	48.79
650	9.467	158.59	980.81	1139.40	0.2786	0.00155	1.5090	0.00337	1.7876	0.00182	39.80
660	11.67	168.69	973.67	1142.36	0.2941	0.00152	1.4753	0.00327	1.7694	0.00175	32.70
670	14.29	178.80	966.53	1145.33	0.3093	0.00150	1.4426	0.00317	1.7519	0.00167	26.99
680	17.37	188.92	959.39	1148.31	0.3243	0.00148	1.4109	0.00308	1.7352	0.00160	22.52
690	20.99	199.06	952.25	1151.31	0.3391	0.00146	1.3801	0.00299	1.7192	0.00153	18.86
700	25.20	209.21	945.10	1154.31	0.3537	0.00144	1.3502	0.00291	1.7039	0.00147	15.88
710	30.08	219.38	937.96	1157.34	0.3681	0.00142	1.3211	0.00283	1.6892	0.00141	13.45
720	35.71	229.56	930.82	1160.38	0.3823	0.00141	1.2928	0.00275	1.6751	0.00134	11.44
730	42.18	239.75	923.67	1163.42	0.3964	0.00139	1.2653	0.00267	1.6617	0.00128	9.792
740	49.57	249.97	916.52	1166.50	0.4103	0.00137	1.2386	0.00260	1.6489	0.00123	8.416
750	57.97	260.19	909.39	1169.58	0.4240	0.00136	1.2126	0.00253	1.6366	0.00117	7.265
760	67.48	270.42	902.25	1172.67	0.4376	0.00134	1.1873	0.00247	1.6249	0.00116	6.298
770	78.21	280.67	895.11	1175.78	0.4510	0.00132	1.1623	0.00241	1.6133	0.00106	5.482
780	90.27	290.94	887.97	1178.91	0.4642	0.00131	1.1385	0.00235	1.6027	0.00104	4.790
790	103.8	301.22	880.83	1182.05	0.4773	0.00130	1.1150	0.00229	1.5923	0.00099	4.201
800	118.8	311.51	873.68	1185.19	0.4903	0.00128	1.0921	0.00223	1.5824	0.00095	3.697
810	135.5	321.81	866.54	1188.35	0.5031	0.00126	1.0698	0.00217	1.5729	0.00091	3.265
820	154.0	332.13	859.40	1191.53	0.5157	0.00125	1.0481	0.00212	1.5638	0.00087	2.894
830	174.4	342.48	852.26	1194.74	0.5282	0.00124	1.0269	0.00207	1.5551	0.00083	2.574
840	196.9	352.83	845.12	1197.95	0.5406	0.00123	1.0062	0.00203	1.5468	0.00080	2.298
850	221.7	363.18	837.97	1201.15	0.5529	0.00121	0.9859	0.00198	1.5388	0.00077	2.061
860	248.8	373.52	830.83	1204.35	0.5650	..	0.9661	..	1.5311	..	1.855

The absolute zero of temperature has been taken as 491.4 deg. Fahr. below freezing-point or −459.4 deg. Fahr.

This expression, or its equivalent, was first deduced by Rankine, and the ideal steam engine, which would convert the whole of this available heat into mechanical work, is said to operate on the Rankine cycle.

The importance of entropy in thermodynamics will be obvious from the foregoing, since a knowledge of it is necessary for determining what fraction of a given amount of heat energy is theoretically capable of conversion into mechanical work. It has, however, no simple physical signification, being merely represented by the integral $\int \frac{dq}{T}$ where dq denotes the heat added to a body at the absolute temperature T.

This integral is due to Kelvin and Rankine, who first recognised its fundamental importance; but the name entropy was subsequently given to it by Clausius, and has come into general use.

No physical change in nature will occur spontaneously unless, as a final result, the total amount of entropy is increased. Such apparent exceptions as may be quoted in opposition to this are always due to artificial conditions, the production of which has at some point or other involved an increase in the total entropy of the universe. So far as it can be discerned for the present, the apparent tendency of nature is to establish an absolute uniformity of conditions throughout all space, and entropy is, in fact, merely the numerical measure of the uniformity attained.

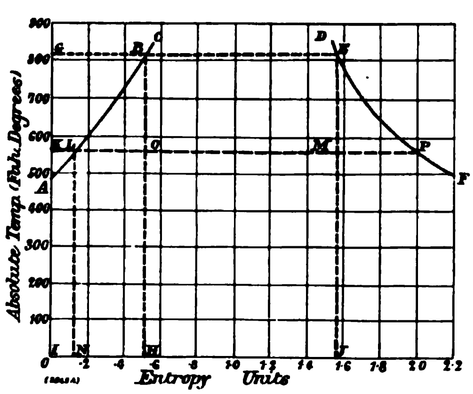


Fig. 61. Temperature Entropy Diagram.

If, taking values from Table XV., we plot down, as in Fig. 61, against the absolute temperature the liquid entropy ϕ_l and the total entropy ϕ of the steam at the different pressures, we get the two curves A B C and D E F, which have some remarkable properties. If we take any point B on the curve A B C, and drop to the base line the perpendicular B H, then the area I A B H is equal to the sensible heat of 1 lb. of steam at the temperature corresponding to B, in this case 820 deg. absolute. That is to say, this area is

equal to 333.13 heat units. The area is measured by multiplying its average height on the temperature scale by the distance I H measured on the entropy scale. If through B the horizontal line G E is drawn to cut the curve D E F at E, then, if the perpendicular E J is drawn, the area B E J H is equal to the latent heat of 1 lb. of steam at this temperature—that is, to 859.40 B.Th.U.

Hence the whole area I A B E J is equal to the total heat of 1 lb. of steam at 820 deg. absolute, or 1191.53 heat units. If, now, another horizontal line K L M is drawn at the temperature 560 deg., say, the area L B E M is equal to the number of units of heat which would be turned into mechanical work in a perfect steam engine of any kind working between these limits of temperature. In other words, the area in question is equal to the available heat, as already defined.

Thus it will be obvious from the diagram that the area L B E M is the sum of the two areas L B O and O B E M. Denoting the higher temperature by T_1 and the lower by T_2, it will be seen that O B E M = $\phi_{s_1} (T_1 - T_2)$, ϕ_{s_1} being the steam entropy at the temperature T_1.

Similarly the area L B O is equal to the whole area I A B H less the two areas I A L N and N L O H; but I A B H is equal to h_1, the sensible heat at the temperature T_1, and A L N I is equal to the sensible heat at the temperature T_2—that is, h_2—whilst the area L O H N is equal to $T_2 (\phi_{l_1} - \phi_{l_2})$, where ϕ_{l_1} = I H, or the liquid entropy at T_1 and ϕ_{l_2} is similarly the liquid entropy at T_2.

Hence the area I A B E J is equal to

$$h_1 - h_2 - T_2 (\phi_{l_1} - \phi_{l_2}) + \phi_{s_1} (T_1 - T_2).$$

which is the value of u as defined in equation (39).

A diagram such as Fig. 61 is known as a temperature-entropy diagram or a $\theta\phi$ chart, and was invented by Willard Gibbs.

From this diagram we can easily prove the simpler value for u given in (40) above.

From Fig. 61 it is evident that the area L B E M is equal to the area A B E J I, less the area A L P I − the rectangle P J.

But, as already stated, the area A B E J I is equal to the total heat in saturated steam at the temperature corresponding to J E or I G, or T_1 say, and this total heat may be denoted by H_1. Similarly the area A L P I is equal to H_2, denoting by this the

total heat of saturated steam at a temperature corresponding to the temperature A K or Q P, or T_2 say. Moreover, the rectangle

$$PJ = PM \times PQ = (KP - GE) \times PQ = (\phi_2 - \phi_1) \cdot T_2 .$$

Hence, finally, the available heat in 1 lb. of saturated steam, when expanded from the pressure corresponding to the temperature T_1 down to that corresponding to the temperature T_2, is $u = H_1 - H_2 + (\phi_2 - \phi_1) T_2$.

Hence if $T_1 = 820°$ and $T_2 = 560°$, we have $H_1 = 1191.53$; $H_2 = 1113.58$; $\phi_2 = 1.9957$; and $\phi_1 = 1.5638$. Whence $u = 319.76$ B.Th.U.

It might be thought that, if we subtracted the total heat H_2 in a pound of steam at temperature T_2 from the total H_1 of the steam at T_1, the difference would be the amount turned into useful work, but it must be borne in mind that there is less than a full pound of steam at the lower temperature, because some of it has been condensed in the process of expansion. In fact the total heat of the steam on exhaust is equal to the area I A M J, of which the portion I A L N is the sensible heat, the remainder being latent. Dry saturated steam at the same temperature has, however, a latent heat equal to the rectangle under the line P L, so that the dryness fraction of the steam on exhaust is given by the ratio

$$\frac{LM}{LP} = \frac{\phi_1 - \phi_{l_2}}{\phi_{s_2}}$$

For the range of temperatures covered in steam practice the value of σ, the specific heat of water, may be expressed as

$$\sigma = 0.5277\ T^{\frac{1}{10}},$$

so that the liquid entropy

$$\phi_l = 5.277\ T^{\frac{1}{10}} - 9.8064, \quad . \quad . \quad . \quad . \quad (41)$$

and the sensible heat

$$h = 0.4798\ T^{1.1} - 437.3.$$

If the steam is not dry to start with, then the equation

$$u = H_1 - H_2 + (\phi_2 - \phi_1)\ T_2$$

still holds if for H_1 we substitute the total heat of the steam in its actual condition, and for ϕ_1 the corresponding entropy.

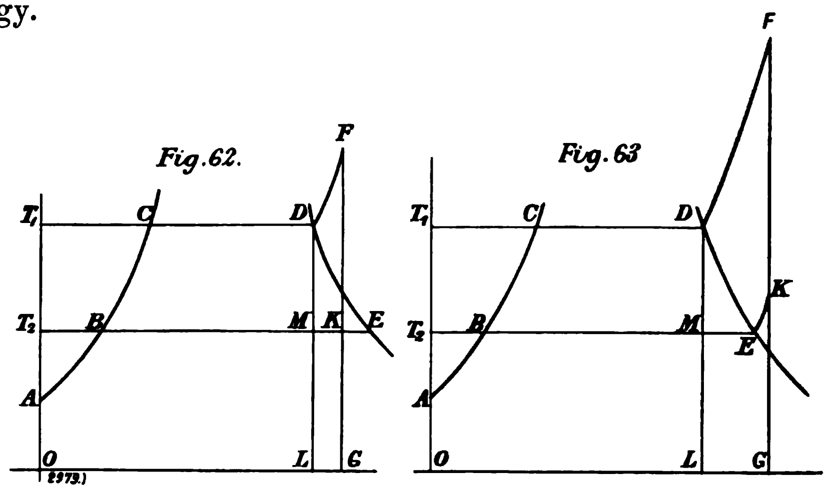
Thus, suppose the temperature conditions as before, but that the steam contains 5 per cent. of moisture, then the actual total heat of the steam is now, if y be the dryness fraction,

$$h_1 + y L_1 = 332.13 + 0.95 \times 859.40 = 1148.56 \text{ B.Th.U.},$$

whilst the actual value of the entropy is similarly

$$\phi_{l_1} + y\,\phi_{s_1} = 0.5157 + 0.95 \times 1.0481 = 1.5114.$$

As before, $H_2 = 1113.58$ and $\phi_2 = 1.9957$, so that we get $u = 1148.56 - 1113.58 + 560\ (1.9957 - 1.5114) = 306.19$ B.Th.U. of available heat as against the 319.65 units which were available when the steam was dry. It will be seen that with the assistance of a steam table the value of u, corresponding to any assumed conditions, can be found without much difficulty, and this represents the maximum amount of energy which, under absolutely ideal conditions, can be converted into mechanical work or into kinetic energy.



Figs. 62 and 63. Temperature Entropy Diagrams with Superheat.

As already explained in Chapter II., knowing u we can find at once the theoretical velocity of efflux of steam from an orifice, which is given by the relation

$$u = 224\sqrt{u}.$$

The temperature entropy diagram represented in Fig. 61 is drawn for saturated steam, but the additions necessary when the steam is superheated are easily made.

Thus in Fig. 62 the point D represents the condition of dry saturated steam at the temperature T_1. By superheating this steam we raise its temperature and increase its entropy, so that its final condition when superheated will be represented by some such point as F. In this case F G denotes the final temperature of the steam, which we may call T_1^1, and O G the corresponding entropy, which may be written as ϕ_1^1. Then the total heat, H_1^1 say, in the superheated steam is equal to the whole area O A C D F G, and the

available heat on expanding it down to the temperature T_2 by the area B C D F K. As before, we get

$$u^1 = H_1^1 - H_2 + T_2 (\phi_2 - \phi_1^1).$$

If u denote the heat available in the case of saturated steam, and u_s the additional amount rendered available by superheating, we have, approximately,

$$u_s = \frac{t}{2}\left(1 - \frac{T_2}{T_1 + \frac{t}{2}}\right).$$

In the foregoing case the expansion has been carried so far that the steam on discharge is wet. When this is not the case matters are a little more complicated. Thus in Fig. 63 the expansion finishes at the point K, where there is still some superheat remaining. The available heat is then represented by the area B C D F K E, and can be found by determining the total heat remaining in the steam in its condition as represented by the point K, and subtracting this from that originally present.

For the purpose of explaining the principles at issue, the temperature entropy diagram has great advantages, but for practical use in turbine designing another form of diagram originated by Professor R. Mollier is more convenient. This is particularly the case when, as in the foregoing instance, the steam is still superheated on discharge.

In the Mollier diagram, Fig. 64, instead of plotting entropy against temperature, entropy is plotted against the total heat in the steam at different pressures and temperatures.

Thus from Fig. 64 it will be seen that steam at 110 lb. pressure and at 500 deg. Fahr. has an entropy of about 1.692, as shown by the vertical scale, whilst from the lower scale it appears that its heat content is 1272 B.Th.U.

With a perfect engine the steam would pass through, doing its work, without change of entropy, since in such an engine no heat is added to the steam by friction, or subtracted from it by radiation or conduction, and the entropy of a body only changes when heat is added to or subtracted from it. Hence suppose that an ideal engine, taking steam under the conditions stated above, is exhausting at 10 lb. pressure. The diagram shows that steam at 10 lb. pressure and an entropy of 1.692 units has a total heat of 1082 B.Th.U.

Hence the perfect engine would have turned into work the difference between 1272 and 1082 B.Th.U., or in other words

$$u = 1272 - 1082 = 190 \text{ B.Th.U.}$$

The quality lines or curves of equal dryness fraction show, moreover, that the steam as exhausted would be 94 per cent dry.

In Chapter V., page 41, it has been shown that if u be known the steam consumption of a theoretically perfect engine is given by the relation

$$w = \frac{2545}{u} \text{ lb. of steam per H.-P.-hour} = \frac{3412}{u} \text{ lb. of steam per Kw.-hour,}$$

and the efficiency ratio was defined as the ratio of this theoretical consumption to the actual steam consumption.

In pressure-compounded turbines the total thermodynamic head is sub-divided up between a succession of relatively small ranges of temperature or pressure. The section of a turbine included between the beginning of one temperature step and the beginning of the next drop in temperature and pressure may be defined as a "stage." Thus in a reaction turbine each row of blades, whether fixed or moving, constitutes a stage, whilst in an impulse turbine the stage always includes not only a set of nozzles and guide blades, but also all other moving or fixed blades through which the steam passes without further fall in pressure. Thus the stage of a Parsons turbine comprises only one friction-producing element, whilst in a velocity-compounded impulse turbine there are frictional losses in the nozzles and in each of the rows of moving and fixed blades.

The "stage" efficiency ratio of a steam turbine is given by the relation

$$\frac{q - f}{q},$$

where q denotes the heat theoretically available in the stage and f the sum of all the frictional losses in the stage.

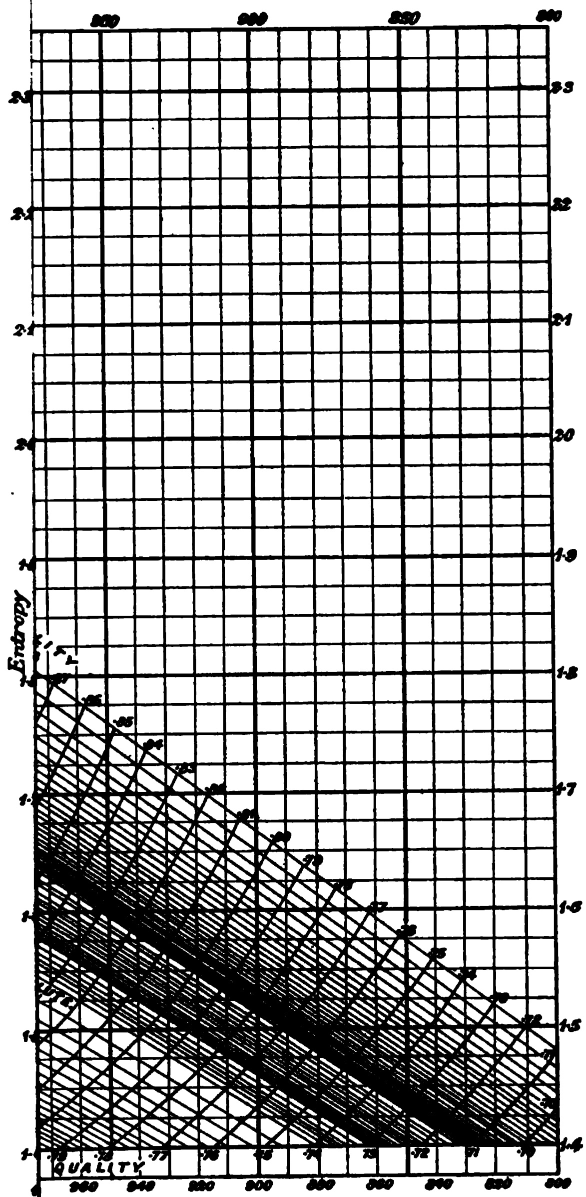
This stage efficiency ratio may for shortness be called the hydraulic efficiency of the turbine, and be denoted by η, so that

$$\eta = \frac{q - f}{q}.$$

There is a close connection between η, the hydraulic efficiency ratio of a turbine, and ϵ its efficiency ratio considered as a whole.

Consider, for simplicity, a turbine in which no energy is carried over in the shape of kinetic energy from one stage to the next.

Fig. 64, PLATE II.



To Face

In that case the whole flow through the guide blades is due solely to the fall of pressure. Let the available heat corresponding to this fall of pressure be q B.Th.U. A portion only of this available heat is actually turned into useful work, the remainder being dissipated in friction and restored to the steam again in the shape of heat. If the fraction thus lost in friction is $k\,q$, then the stage efficiency is

$$\eta = \frac{(1-k)\,q}{q}.$$

Now in all cases where steam is expanded down to a lower pressure in a non-conducting vessel, the total heat theoretically available always appears in the first instance as high-grade energy, being represented by the external work done added to the kinetic energy of such jets and eddies as may be formed. The kinetic energy of the latter is finally dissipated in fluid friction, or, in other words, is restored to the steam in the shape of heat. Hence, if the hydraulic efficiency of a stage is

$$\eta = \frac{(1-k)\,q}{q},$$

$k\,q$ represents energy, which during the expansion in the stage has been added to the steam in the shape of heat. Hence in passing through the stage the steam has its entropy increased by the amount

$$\frac{k\,q}{\mathrm{T}},$$

where T denotes the absolute temperature of the steam in the stage.

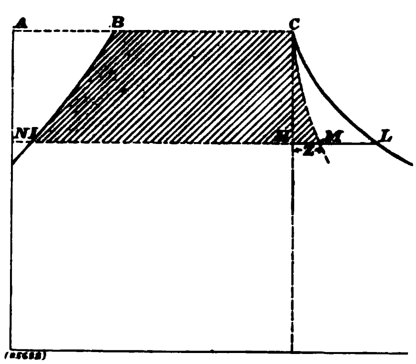


Fig. 65. Effect of Reheat in Temperature Entropy Diagram.

As a consequence, in the temperature entropy diagram, Fig. 65, the expansion line of the steam passing through a multi-stage turbine is in practice represented by the curved line C M in place of by the vertical line C H, which is the expansion line corresponding to perfect efficiency. Hence at the temperature corresponding to the line N L the dryness fraction of the steam on delivery from the corresponding stage is equal to $\frac{\mathrm{I\,M}}{\mathrm{I\,L}}$ in place of $\frac{\mathrm{I\,H}}{\mathrm{I\,L}}$, which would be its value in the case of a perfect turbine.

The equation to the line CM can be calculated in the case of a turbine having an infinite number of stages, and for turbines of more than some six or seven stages the limiting points of the actual expansion may, with sufficient accuracy, be considered as lying in the curve thus found.

Thus let the point M, Fig. 65, represent the condition of the steam in the stage, having a temperature T, of a turbine with an infinite number of stages. If the drop of temperature to the next stage is $d\,T$, then, having regard to the sign of T, the heat available for doing work there is $IM \times -d\,T$. Hence, if ϕ_1 denote the initial temperature of the steam, and ϕ_l its liquid entropy at the temperature T, we have

$$\begin{aligned} d\,q &= IM \times -d\,T \\ &= -(\phi_1 - \phi_l + Z)\,.\,d\,T, \end{aligned}$$

where ϕ_l is equal to IN, and Z = HM.

Of this available heat let the fraction $k\,d\,u$ be wasted in friction and restored to the steam as heat, so that at the next stage the entropy is increased by the amount $d\,Z = \frac{k\,d\,q}{T}$.

We thus have

$$T\,d\,Z = -k(\phi_1 - \phi_l + Z)\,d\,T$$

or

$$T\frac{d\,Z}{d\,T} + k\,Z = -k(\phi_1 - \phi_l).$$

Multiplying by T^{k-1} this gives

$$T^k\,.\,\frac{d\,Z}{d\,T} + k\,Z\,.\,T^{k-1} = -k(\phi_1 - \phi_l)\,T^{k-1},$$

i.e.,

$$\frac{d}{d\,T}\,.\,(Z\,.\,T^k) = -k(\phi_1 - \phi_l)\,T^{k-1}$$

or

$$Z\,T^k = A - \phi_1\,T^k + k\int\phi_l\,T^{k-1}\,d\,T,$$

where A is the constant of integration.

Substituting form (41), page 145,

$$\phi_l = 5.277\,T^{\frac{1}{10}} - 9.8064$$

we get

$$Z = \frac{A}{T^k} - \phi_1 - \frac{9.8064}{10\,k + 1} + \frac{10\,k}{10\,k + 1}\,.\,\phi_l$$

The constant A is determined by the fact that when $T = T_1$ $Z = 0$.

In the diagram, Fig. 65, the heat which becomes available as

the steam passes through the turbine is equal to the whole hatched area, whilst that which is available in a perfect turbine is equal to B C H I only. Hence the reheat factor R is given by

$$R = \frac{BCMI}{BICH} . = 1 + \frac{CHM}{BCHI}.$$

The area B C H I is equal to u, whilst the area C H M is equal to

$$\int_{T_2}^{T_1} Z\,d\mathrm{T}.$$

Substituting from the expression found above for Z, we get

$$\int_{T_2}^{T_1} Z\,d\mathrm{T} = \frac{A}{1-k}\left(T_1^{1-k} - T_2^{1-k}\right) + \frac{100\,k}{k+1}\left(h_1 + h_2\right) - (\phi_1 + 9.8064)(T_1 - T_2).$$

In this way, for steam expanded from an absolute temperature of 820 deg. (equivalent to nearly 154 lb. absolute) down to 560 deg. (equivalent to about a 28-in. vacuum), we get the following values showing the relation between the hydraulic efficiency and the efficiency ratio of a turbine with an infinite number of stages :—

$\eta = 1$	0.9	0.8	0.7	0.6	0.5
$\epsilon = 1$	0 916	0.8286	0.7378	0.6432	0.5455.

The corresponding reheat factor is equal to $\frac{\epsilon}{\eta}$.

Another method of determining these reheat factors is based upon the use of the approximate equation $p\,V^\gamma$ = constant, where the value of γ changes with the efficiency of the turbine, and can be determined for any conditions by the method explained in Chapter II., page 13.

Thus let γ_1 be the value of γ corresponding to adiabatic expansion. Then we have

$$u = \frac{144}{778} \cdot \frac{\gamma_1}{\gamma_1 - 1} \cdot p_0 V_0 \left(1 - \frac{\rho_1}{x}\right).$$

If the efficiency ratio however be, say, 0.7, then we can, from the Mollier diagram, determine the corresponding dryness fraction of the steam on exhaust, and from this we obtain $\rho_{0.7}$, the corresponding ratio of expansion, and we thus get $\gamma_{0.7}$, the corresponding value of γ. The work now done by the steam is equal to, say, U; B.Th.U. where

$$U = \frac{144}{778}\ \frac{\gamma_{0.7}}{\gamma_{0.7} - 1} \cdot p_0 V_0 \left(1 - \frac{\rho_{0.7}}{x}\right).$$

The reheat factor is then given by the relation

$$R_{.7} = \frac{U}{u} = \frac{\gamma_{0.7}}{\gamma_1} \cdot \frac{\gamma_{0.7} - 1}{\gamma_1 - 1} \cdot \frac{x - \rho_{0.7}}{x - \rho_1}.$$

In this way the curves, Fig. 35, page 44, showing the relation between values of η and ϵ for different values of x, have been determined, as well as the table of reheat factors printed on page 45, *ante*. The curves, Fig. 35, show that once the value of x attains a certain value the reheat factor alters very little.

Tables giving the volume of superheated steam are not common. The formula $p\,V = c\,T$, which is very nearly true for permanent gases, is not accurate if applied to a condensible vapour such as steam. In fact for all actual gases the above equation should in strictness be replaced by the expression

$$(p + \pi)(V - b) = c\,T,$$

where π denotes the attraction of the molecules for each other, which becomes sensible when the gas is greatly condensed. It is this attraction which appears as capillary attraction and as the surface tension of liquids. The term b denotes the volume of 1 lb. of the molecules, if the whole were absolutely in contact with each other. For water $b = 0.014$ cub. ft. approximately.

With some fluids π can be expressed in the simple form $\frac{a}{V^2}$, where a is a constant; but this is not true in the case of steam, for which Callendar's researches give the following relation

$$V - 0.014 = \frac{0.593\,T}{p} - 1.5\left[\frac{459.4}{T}\right]^{\frac{10}{3}}.$$

where p is in pounds per square inch and T is taken in Fahrenheit degrees. For many purposes, however, the volume of superheated steam is given with sufficient accuracy by the relation

$$V^1 = V \cdot \frac{T^1}{T}$$

where V^1 denotes the volume of 1 lb. of the steam at the absolute temperature T^1, and V denotes the volume of 1 lb. at its absolute temperature of saturation T.

CHAPTER XVII.

BALANCING.

THE adoption of the steam turbine has introduced an entirely new order of ideas as to what constitutes a properly-balanced driving-shaft. A defect from perfection in this regard, which would have been of quite negligible importance in the case of a slow-speed reciprocating engine, would suffice to wreck a rapidly-revolving turbine rotor. The methods used to secure the accurate balance of turbine rotors have, in the main, been developed by the Hon. Sir C. A. Parsons.

The operation of balancing a rotor is effected in two successive stages. The first of these is directed to securing static balance. To this end the rotor is laid upon accurately-levelled knife edges, and temporary balance weights applied to it, until it will rest in neutral equilibrium, whatever position it occupies. A common method of effecting this is to divide the circumference of the rotor into six equal arcs, and to turn it until each point of division is successively brought to the uppermost position. In general in each position a balance weight must be added to one side or the other to prevent the rotor from turning. The weight added in each position shows the amount the rotor is out of balance in that position. On tabulating these numbers it is easy to ascertain the exact extent (both in amount and direction) to which the centre of gravity of the rotor lies outside of the axis of figure, and to determine the precise position and amount of the balance weights to be added to correct this. Mr. J. M. Newton, in his paper (Proceedings of the Institution of Junior Engineers, vol. xx.) states that with a 2-ton rotor this adjustment is continued until the centre of gravity does not lie outside the axis by more than 0.0005 in. to 0.0007 in. The residual amount of out of balance is then (in the case of a 2-ton rotor) equivalent to a weight of $\frac{1}{4}$ lb. acting at a 12-in. radius.

In the case of disc rotors, the discs or wheels are balanced separately by methods equivalent to the foregoing before mounting

them on the shaft. Some builders, for balancing these discs, use a regular weighing machine. This carries as a prolongation of its axis of suspension a mandril taking the wheel, and on this mandril the wheel can be clamped in any one of six positions, and balanced by adding weights to one or other of the scale pans. With a perfectly-balanced wheel no weight should be required in either scale pan, whatever the position of the wheel on its mandril. Actually, however, as the latter is turned successively into each of its six positions, the weight has to be adjusted, and the tabulated figures give data by which the departure of the centre of gravity of the wheel from its designed position can be calculated, and subsequently corrected. Theoretically, for the making of this calculation it should be sufficient to measure the want of balance in two positions only, which should by preference be at right angles to each other.

After static balance has been thus secured, which, in the case of drum rotors, is effected by adding temporary weights where and as required, the rotor has to be tested for dynamic balance. To this end it is mounted on two bearings resting on ball bearings and kept in position by adjustable springs. In these bearings it is driven round by a suitable motor and through a flexible coupling at a speed which ultimately is made, in the case of a drum rotor, about 20 per cent. higher than that at which it is intended to run in service. In the earlier stages of the operation the two supporting bearings are best clamped solid. Starting the rotor into slow rotation, a number of sections are found at which a coloured pencil, held lightly against the rotating surface, marks it all the way round. The speed is then raised. Now, any body rotating at a high speed endeavours to move so that its centre of gravity lies on the axis of rotation, and, secondly, so that this axis is such that the moment of inertia of the rotor about it is either a maximum or a minimum. The former condition is that "preferred" by the body. Thus if a stout disc of cardboard is supported at the edge by a string, it will be found that on rotating the upper end of the latter the disc, so long as the rotation is slow, will rotate about its vertical diameter. As the speed is increased, however, it tends to become horizontal, and at very high speeds turns horizontally, with its centre very near to the axis of rotation in spite of the side pull due to the fact that the supporting string is fixed near one edge.

A long rotor can therefore rotate in balance either about one longitudinal axis or an indefinite number of transverse axes. If set rotating about any other longitudinal axis, a controlling force is needed to maintain this condition, and such forces are supplied by the bearings. If the speed be high, and the weight great, these forces, which are alternating, or, perhaps one should say, rotatory in character, are considerable, and set up serious vibrations. If, then, a rotor is mounted on bearings and rotated as described above, it will, if out of balance, apply forces to the bearings, and shift them to and fro so as to try and get its centre of gravity into the axis of rotation, and to get one of its "principal" axis coincident with the latter.

Hence a coloured pencil held against the same part of the rotor as that it touched all round at low speeds, will, at high speeds, touch it over a short arc only. To bring the surface true again, weights must be added to this "high side," so as to shift the centre of gravity in the same direction. Actually the "high parts" at opposite ends of the rotor will generally be on opposite sides, this being due to the fact that the actual axis of rotation is not a principal axis. By adding weights to the respective "high sides" the position of this principal axis can be adjusted to practical coincidence with the axis of rotation. It is sometimes found that the balance-weight requires to be added about 120 deg. in advance of the point not marked by the pencil instead of 180 deg. in advance, as required by theory. When the operation of balancing is nearly completed, the pencil marks nearly all round, and it becomes difficult to determine where further weights should be added, and to what extent. The delicacy of the coloured pencil test can then be increased by adjusting the speed so that the springs which control the bearings are in resonance with the rate of revolution at which the rotor is driven.

The attainment of dynamic balance is not always sufficient to secure the steady running of a rotor, it being further essential that the usual running speed shall not lie near a certain critical value. How such a critical speed arises may be realised from the following considerations: Take, for instance, the case of a rotor of a horizontal turbine of the wheel type. To each wheel affix a weight equal to the weight of the wheel, thus doubling the deflection of the shaft. If, now, all these additional weights are suddenly cut loose, the shaft

will fly back past its normal position and continue to vibrate about the latter for some time. The number of vibrations made per second will obviously depend both on the rigidity of the staff and on the weight ot the wheels it carries. If the shaft be very rigid and the wheels light, the periodicity of the vibration will be high, and, conversely, a rotor consisting of a flexible shaft and heavy wheels has a slow natural rate of vibration. Whatever this natural period of vibration may be, it is easy to understand that, should the rate of revolution coincide with this rate, resonance effects may ensue, augmenting the range of oscillation beyond all limit.

The matter may also be viewed from another standpoint. In actual practice it is not possible to make the centre of gravity of a rotor absolutely coincident with the axis of rotation. Thus in Fig. 66 let A B represent an elastic shaft, which for simplicity will be considered weightless. Let it carry a weight W, the centre of gravity of which is distant Δ inches from the axis of rotation.

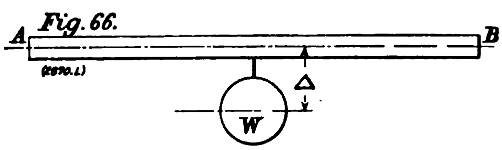


Fig. 66. Unbalanced Shaft.

If, then, the shaft be set in rotation a centrifugal force will be developed which will deflect the shaft by an amount equal to δ inches, so that the distance of the centre of gravity from the axis of rotation becomes equal to $\Delta + \delta$. The corresponding centrifugal force will be equal to $\frac{W}{g} \cdot \frac{(\Delta + \delta)\,\omega^2}{12}$ where ω denotes the angular velocity of rotation, *i.e.*, $\omega = \frac{\pi N}{30}$ where N = R. P. M. A deflected shaft is essentially a spring, and opposes to the centrifugal force a counterbalancing force, which is equal to $f\delta$ where f denotes a coefficient depending solely on the type and dimensions of the shaft and on the position of the load.

For a uniform shaft centrally loaded the value of the resisting force F is

$$F = \frac{2.236\, D^4 . E}{l^3}\, \delta.$$

where D denotes the diameter of the shaft in inches. When the weight is not central, but situated at a distance $k\,l$ from the nearest support, the value of F may be written as $F = \frac{D^4 E}{c\, l^3}$, where the

value of c is taken from the curve given in Fig. 67. In this figure $k = \frac{a}{l}$, where a denotes the distance of the load from the nearest support, and l the span.

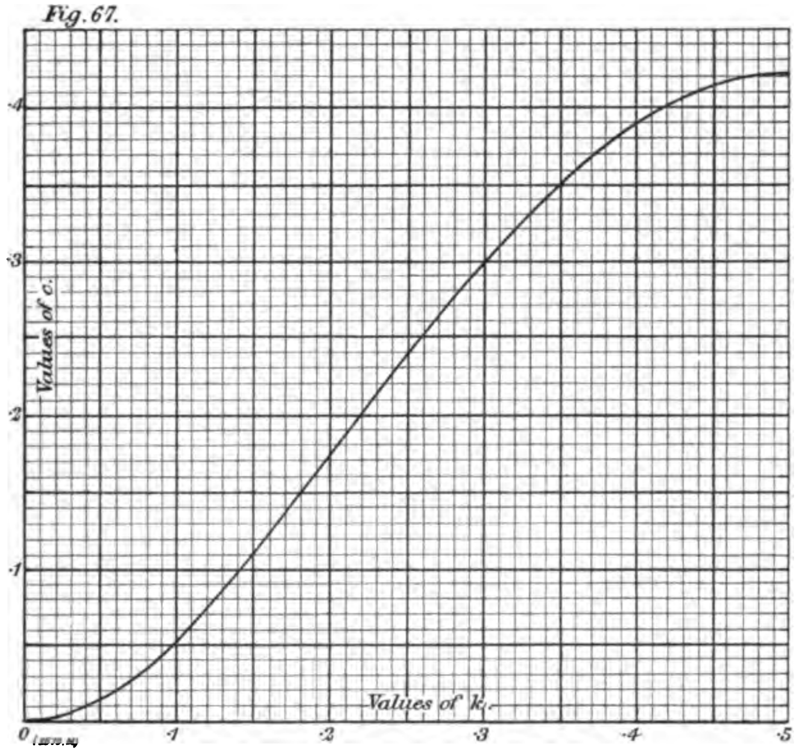


Fig. 67. Diagram for Calculating Critical Speeds.

In the case of a rotating shaft the deflection increases until this counterbalancing force is equal to the centrifugal force. We then get

$$F = \frac{D^4 E}{c\, l^3} \delta = \frac{W}{g} \cdot \frac{(\Delta + \delta)\, \omega^2}{12}.$$

Whence

$$\delta = \frac{\frac{W}{g} \cdot \frac{\Delta\, \omega^2}{12}}{\frac{D^4 E}{c\, l^3} - \frac{W}{g}\, \frac{\omega^2}{12}}$$

Now the first term in the denominator is constant for a given shaft and a given position of the load, whilst the second, being dependent on the square of the angular velocity, increases as the speed rises. The critical speed, then, is given by the condition that

$$\frac{D^4 E}{c\, l^3} = \frac{W}{g} \cdot \frac{\omega^2}{12} \quad . \quad . \quad . \quad . \quad . \quad . \quad (42)$$

If the speed exceeds this value, the deflection, it will be seen, becomes negative, and with ω = infinity, we get $\delta = -\Delta$, or, in other words, the shaft has then deflected, so that the centre of gravity has been brought into coincidence with the axis of rotation.

A well-balanced rotor will run steadily either above or below its critical speed, but may vibrate very badly at this critical speed. The practice of running rotors above their critical speeds seems to have been introduced by Dr. de Laval.

From the foregoing equation (42) the critical speed for a uniform weightless shaft carrying a heavy wheel in any position can be calculated.

Let this critical speed be denoted by ω_1. Suppose this wheel is removed and a second added in a different position, then the new critical speed will be different, say, ω_2. Similarly, for a third wheel, the critical speed thus calculated may be taken as ω_3, and so on. Then, if all the wheels are in place at the same time, the critical angular velocity for the whole system thus constituted is given by Dunkerley's equation, viz.—

$$\frac{1}{\omega^2} = \frac{1}{\omega_1^2} + \frac{1}{\omega_2^2} + \frac{1}{\omega_3^2} \text{ \&c.}$$

Further, since the unloaded shaft is not actually weightless, as was provisionally assumed, it also has a critical speed, ω_0, which is given by the relation $\frac{1}{\omega^2} = \frac{0.209 \text{ W}_0 l^3}{12 g . \text{D}^4 . \text{E}}$, where W_0 denotes the weight of the unloaded shaft. So that finally we get for the critical speed

$$\frac{1}{\omega^2} = \frac{l^3}{12 d^4 \text{E} g}\left\{\text{W}_1 c_1 + \text{W}_2 c_2 + \text{W}_3 c_3 + \text{\&c.}, + 0.209 \text{ W}_0\right\},$$

where the values of c are read from Fig. 67, and W_0 is the total weight of the shaft itself.

Thus, suppose a 6-in. shaft, 60 in. long between centres of bearings, and carrying two wheels each weighing 500 lb., the first being mounted at 20 in. from the left-hand bearing and the other at 29 in. from the right-hand bearing. Then for the first wheel the value of k is $\frac{20}{60} = 0.333$, and from Fig. 66 the corresponding value of c is 0.335. In the second wheel the value of k is $\frac{29}{60} = 0.484$, so that c is equal to 0.423. The weight of the shaft is equal to 480 lb.,

so that $W_0 \times 0.209 = 100$, whence

$$\frac{1}{\omega^2} = \frac{l^3}{12 D^4 E . g} . \left[500 \times 0.335 + 500 \times 0.423 + 100\right] = \frac{60^3 \times 479}{12 \times 6^4 \times 30{,}000{,}000 \times 32.2},$$

$\therefore \omega = 381$ radians per second or 3639 turns per minute.

This is the critical value for a shaft supported at both ends, and may be denoted by ω_c. For other conditions the actual critical speed will be equal to $\phi . \omega_c$, where the value of ϕ is to be taken from the following table:—

Method of Supporting Shaft.	ϕ.
Cantilever	0.3563
One end fixed, the other supported	1.560
Both ends fixed	2.250

In the case of certain Continental turbines, the turbine and the generator shaft are made continuous, the whole being supported on three bearings. With this plan it was imagined that the conditions of running would be equivalent to one end of each of the two sections of the shaft being supported and the other fixed, so that the critical speed would be increased. Unfortunately, however, the designer overlooked the fact that the two sections of the shaft could bend in opposite directions, the stiffness being then the same as if each were simply supported. The system of construction is cheap, but there have been some serious difficulties with it in practice, and in the early days there were many cases of bent and broken shafts. In the case of two large units of 7500 kilowatts each, moreover, the turbines could not be run at more than one-third of their rated output until entirely reconstructed. The trouble apparently arose here from distortion of the casing, as with high loads the superheat extended further towards the exhaust end. This presumably threw the bearings out of line. There is still occasional trouble with these two turbines, from vibration when starting them up after a stoppage. Hence the plan is one which should be adopted with caution.

Where the moment of inertia of a turbine shaft varies greatly, the critical speed is more conveniently found in another way than that explained above. Suppose the rotor in question to be placed horizontally, then under its own weight it will be deflected, and the amount of the deflection at any point can be calculated. The actual load on the rotor is, of course, a continuous one, but for practical

purposes it can be replaced by a number of isolated loads, p_1, p_2, p_3, &c., such as indicated in Fig. 68.

If δ_1 be the deflection of the shaft at the point occupied by p_1, δ_2 the corresponding deflection for p_2, and so on, the potential energy stored in the deflected rotor is then equal to

$$\frac{1}{2}\left[p_1\,\delta_1 + p_2\,\delta_2 + p_3\,\delta_3 + \&c.\right]$$

$$= \frac{1}{2}\sum_1^n p_n\,\delta_n.$$

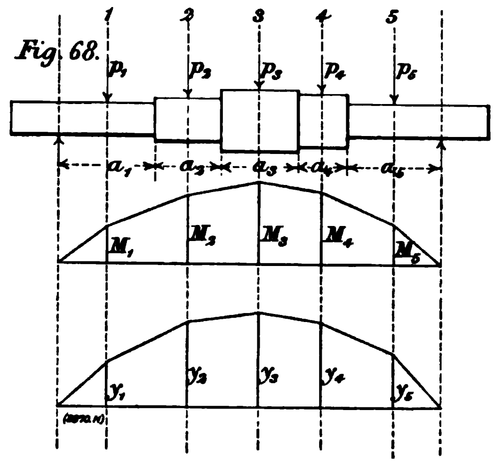


Fig. 68. Graphic Calculation of Critical Speeds.

Let, next, the rotor be placed with its shaft vertical. It will then straighten itself, but it can be bent to the same curve as before by applying horizontal forces equal to the loads p_1, p_2, &c. Assume this to be done, then the potential energy stored in the bent rotor will, as before, be equal to $\frac{1}{2}\sum_1^n p_n\ \delta_n$.

If, now, these horizontal forces be suddenly cancelled, the rotor will fly back and begin to oscillate with its natural period of vibration, and, as already stated, the total number of complete vibrations per second will be equal to the number of revolutions per second at the critical speed.

At the mid point of each oscillation the centre line of the shaft is straight, and hence there is at this instant no potential energy stored in it, the whole of the energy maintaining the vibration being then kinetic. If N denotes the number of complete cycles made per second, then as each weight vibrates about its neutral position according to the law $\delta = \delta_0 \sin \frac{\pi N}{30} t$, its velocity of motion at any instant is equal to

$$\frac{d\,\delta}{d\,t} = \delta_0 \cdot \frac{\pi N}{30} \cdot \cos \frac{\pi \cdot N}{30} t,$$

which, when $\delta = 0$, becomes

$$\frac{d\,\delta}{d\,t} = \delta_0 \frac{\pi N}{30}.$$

The kinetic energy of any weight p_n as it passes through its

neutral position is, therefore, equal to $\frac{p_n}{2g} \cdot \frac{\delta_n^2}{144} \cdot \frac{\pi^2 N^2}{900}$ ft.-lb. or $\frac{1}{24g} \cdot \frac{\pi^2 N^2}{900} \cdot p_n \delta_n^2$ in.-lb., so that the total kinetic energy of the whole of the moving weights at the instant the shaft is straight is equal to

$$\frac{1}{24g} \cdot \frac{\pi^2 N^2}{900} \left\{ p_1 \delta_1^2 + p_2 \delta_n^2 + \&c. \right\} \text{ in.-lb.}$$

and this must be equal to the potential energy of the strain already found, whence

$$\frac{1}{24g} \cdot \frac{\pi^2 N^2}{900} \cdot \sum_1^n p \delta^2 = \frac{1}{2} \sum_1^n p \delta.$$

From this we get the relation

$$\frac{\pi^2 N^2}{900} = \omega^2 = \frac{12 g \Sigma p \delta}{\Sigma p \delta^2}.$$

To determine the values of δ we proceed as follows:—

First draw the bending moment curve as indicated in Fig. 68. Find the ordinates at the points 1, 2, 3, &c., and let these be denoted by M_1, M_2, M_3, &c. Then, with imaginary loads equal to

$$\frac{M_1 a_1}{d_1^4} \cdot \frac{M_2 a_2}{d_2^4}, \&c.,$$

where d_1 denotes the diameter of the shaft at p_1, and so on, draw a second curve of imaginary bending moments; let these be denoted by y_1, y_2, y_3, &c.

Then

$$\delta_1 = \frac{64}{E\pi} y_1 \qquad \delta_2 = \frac{64}{E\pi} y_2, \text{ and so on.}$$

Hence

$$\omega^2 = \frac{12 g E \pi}{64} \cdot \frac{\Sigma p y}{\Sigma p y^2}.$$

Taking E as 30,000,000, we get

$$\omega = 23{,}850 \sqrt{\frac{\Sigma p y}{\Sigma p y^2}}.$$

The calculation of the deflections can, of course, also be effected arithmetically, and in Table XVI. we give the results obtained for the rotor of the Westinghouse-Rateau 5000-kw. turbine, illustrated in another Chapter. Some of the dimensions have had to be scaled from the engravings, and may thus not be absolutely exact.

TABLE XVI.—CALCULATION OF CRITICAL SPEED OF ROTOR FOR 5000-KW. TURBINE.

Section Number.	Length of Section a in Inches.	Loads in Pounds.	Bending Moment M in Pounds.	Imaginary Load $\frac{M\,a}{d^4}$.	$\frac{p\,y}{10,000}$.	$p\left[\frac{y}{10,000}\right]^2$.
1	$19\frac{7}{8}$	741.3	240,860	12,798	949	12,142
2	$4\frac{5}{8}$	133.1	536,665	25,774	343	884
3	$17\frac{3}{4}$	974.1	798,930	36,488	3,554	12,969
4	$1\frac{5}{8}$	183.8	1,023,575	42,819	787	3,370
5	$6\frac{5}{8}$	2124	1,112,250	44,505	9,453	42,070
6	6	2071	1,241,850	47,592	9,856	46,908
7	$5\frac{3}{4}$	2064	1,350,050	50,065	10,333	51,734
8	$4\frac{3}{4}$	2011	1,437,800	51,927	10,443	54,225
9	$5\frac{1}{4}$	2044	1,510,450	53,429	10,900	58,237
10	$3\frac{3}{4}$	1916	1,566,600	54,514	10,445	56,939
11	$4\frac{3}{8}$	1975	1,608,350	55,326	10,927	60,454
12	$4\frac{3}{8}$	1971	1,644,700	55,997	11,037	61,804
13	6	2236	1,676,350	56,562	12,647	71,534
14	4	2062	1,698,550	56,816	11,715	66,560
15	4	2073	1,707,100	56,872	11,790	67,052
16	$1\frac{3}{4}$	207	1,709,150	56,810	1,176	6,681
17	4	2078	1,707,075	56,719	11,786	66,849
18	4	2063	1,698,825	56,453	11,646	65,766
19	4	2044	1,683,650	56,043	11,455	64,197
20	4	2031	1,661,600	55,476	11,267	62,505
21	4	2016	1,631,350	54,745	11,036	60,416
22	4	2002	1,593,000	53,843	10,779	58,037
23	4	1989	1,546,450	52,762	10,494	55,368
24	4	1975	1,491,950	51,496	10,170	52,371
25	4	1962	1,429,750	50,040	9,818	49,128
26	4	1949	1,359,750	48,388	9,431	45,734
27	4	1936	1,281,900	46,539	9,010	41,932
28	$3\frac{7}{8}$	1924	1,197,050	44,521	8,566	38,137
29	$3\frac{7}{8}$	1899	1,106,000	42,340	8,040	34,041
30	2	208	1,033,500	40,554	843	3,421
31	$13\frac{3}{4}$	582	829,480	35,504	2,066	7,336
32	$4\frac{1}{2}$	151	592,090	26,742	404	1,080
33	$20\frac{1}{8}$	648	266,660	13,461	872	1,174

Adding up the last two columns of the table we get

$$\Sigma p \left[\frac{y}{10{,}000}\right] = 264{,}038$$

and

$$\Sigma p \left[\frac{y}{10{,}000}\right]^2 = 1{,}381{,}055.$$

Whence

$$\frac{\Sigma p y}{\Sigma p y^2} = \frac{1}{52{,}302},$$

and

$$\omega = 23{,}850 \sqrt{\frac{1}{52{,}302}} = \frac{23{,}850}{228.7} = 104.3.$$

Whence

$$\mathrm{N} = \frac{30\,\omega}{\pi} = 995.8 \text{ revolutions per minute.}$$

This, it will be seen, is not very greatly in excess of the designed running speed, and illustrates the difficulty of securing a really stiff rotor with the disc type of machine.

It is therefore not uncommon to run such machines above their critical speeds. In that case, however, the efficiency of the turbine may be expected to diminish with time, since on every occasion on which the turbine is started up or stopped the rotor has to pass through its critical speed, and the consequent vibration gradually enlarges the fine clearances used where the shaft passes through the high-pressure diaphragms.

A fair approximation to the critical speed of a rotor can be obtained by replacing the actual shaft by an ideal shaft of the same stiffness but of uniform diameter throughout, and the actual load by an ideal uniformly-distributed load producing the same maximum bending moment.

In that case the critical speed is given by the relations

$$\omega = 83{,}280\,\frac{d^2}{l}\cdot\frac{1}{\sqrt{\mathrm{M}}} \quad . \quad . \quad . \quad . \quad . \quad (43)$$

or

$$\mathrm{N} = 795{,}000\,\frac{d}{l}\cdot\frac{1}{\sqrt{\mathrm{M}}} \quad . \quad . \quad . \quad . \quad . \quad (44)$$

where d denotes the diameter in inches of the equivalent shaft, l its length between centres of bearings, and M the maximum bending moment in inch-pounds caused by the actual load.

To determine d, the diameter of the equivalent shaft, the span of the actual shaft is divided into four equal parts. Numbering the

points of section 0, 1, 2, 3, and 4, and letting d_0, d_1, d_2, d_3, and d_4 be the corresponding diameters, we have for the value of d the relation

$$[d]^4 = \frac{7\,(d_0^4 + d_4^4) + 32\,(d_1^4 + d_3^4) + 12\,d_2^4}{90}.$$

When the shaft is of fairly symmetrical proportions, this may be replaced by the simpler formula

$$d = \frac{7\,(d_0 + d_4) + 32\,(d_1 + d_3) + 12\,d_2}{90}.$$

To apply this rule to the 5000-kilowatt rotor, of which the critical speed has been calculated above, we note that the maximum bending moment M is about 1,709,000 in.-lb. We also have

$$d_0^4 = 20{,}732\,;\ d_1^4 = 102{,}070\,;\ d_2^4 = 194{,}480\,;\ d_3^4 = 102{,}070\,;\ \text{and } d_4^4 = 20{,}732.$$

Hence

$$d^4 = 101{,}744 \text{ and } d^2 = 308.8.$$

Substituting these values in (43) and (44) gives

$$\omega = 102.1 \text{ radians per second,}$$

or

$$N = 975.4 \text{ revolutions per minute.}$$

These values are nearly the same as was found by the detailed calculation already given.

When the shaft is hollow the formula for the equivalent diameter is

$$d^4 = \frac{7\,(D_0^4 + D_4^4 - d_0^4 - d_4^4) + 32\,(D_1^4 + D_3^4 - d_1^4 - d_3^4) + 12\,(D_2^4 - d_2^4)}{90},$$

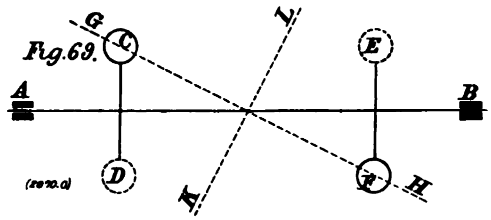
where D denotes the outer and d the inner diameter at each section.

In using the above rules for finding the diameter of an equivalent shaft, a certain amount of intelligence must be exercised. The diameters d_1, d_2, d_3, &c., should be fairly representative of the general character of the shaft in their neighbourhood.

In the foregoing treatment of critical speeds it has been implicitly assumed that the various wheels, &c., mounted on the shaft simply oscillate to and fro in a straight line, when the shaft is vibrating naturally. This is true for wheels near the centre of the shaft, but those near the ends also vibrate through a small angle, as the shaft bends in and out. This, in effect, increases the inertia of these weights, and thus reduces the critical speed. The correction is, however, nearly always insignificant from a practical standpoint. Those who desire to study this point further will find a pretty complete treatment of it by Professor A. Morley in "ENGINEERING," vol lxxxviii., page 135, *et seq.*

It was stated on page 154 that a body would only rotate quietly about one of its principal axes, and would by preference chose that axis about which its moment of inertia was a maximum. The principle involved in this statement will be clear on reference to Fig. 69, where A B denotes a weightless shaft carrying at the ends of equal weightless arms the four weights C D E F. With this arrangement the system will rotate quietly about the axis A B, which from the symmetry of the conditions is one of the principal axes. If, now, the D and E be removed, leaving C and F in place, A B, though it still passes through the centre of gravity of the system, is no longer a principal axis, and if an attempt is made to establish rotation about this line the centrifugal forces developed will obviously tend to shift the axis of rotation into the line K L, which under the new conditions is the principal axis about which the moment of inertia of the system is the greatest. Another principal axis (when D and E are removed) is represented by the line G H, and quiet rotation can also be maintained around this. If, however, from any accidental cause the axis of rotation is caused to diverge slightly from G H, it will be obvious from the figure that this divergence will tend to increase owing to the centrifugal forces then developed, whilst when rotating about K L any accidental temporary displacement of the axis of rotation will tend to be automatically eliminated by the ensuing centrifugal forces.

Fig. 69.



CHAPTER XVIII.

DUMMY AND GLAND PACKINGS.

IN each group of a Parsons turbine the steam besides exerting a torque on the drum also produces an axial pressure on the blades, the amount of which is equal to half the pressure difference between the beginning and end of the group multiplied by the total annular area between the drum and the casing at that group. To prevent the drum being forced endways by this pressure dummy pistons are provided. These cannot be made a close fit with the casing, as at the very high speed of rotation the friction developed would almost instantly lead to "seizing." At the same time it is essential that there shall be no serious leakage of steam past them, as such leakage effects no useful work and constitutes a dead loss.

To meet the need for complete freedom from friction, combined with sufficient steam tightness, the so-called labyrinth packing was devised by Sir C. A. Parsons. The term "labyrinth packing" is not well chosen, as the essential idea on which the device is based is not so much the causing of the steam to take a tortuous path as to wiredraw it at a great number of points.

The character of the packing is well shown in Fig. 70, page 171. The rings on the casing are usually of brass, though steel is used in some cases where the steam temperature is very high. They are commonly $\frac{3}{16}$ in. thick, and are set at a pitch of $\frac{1}{2}$ in. or $\frac{3}{8}$ in. They engage, it will be seen, with grooves turned on the rotor, and the position of the latter is adjusted axially, so as to bring each brass ring almost into contact with one face of a groove on the rotor. The two adjacent faces are, in fact, ground and scraped together, and then finally the rotor is shifted axially, so as to allow a distance of a few mils between the faces of the opposing rings. With brass rings a curious phenomenon is often observed after the turbine has been at work for some time. The

rings are sometimes inserted in short 6-in. to 8-in. lengths, which, as originally fixed, abut one against the other. After being in use for some time, however, gaps of $\frac{1}{4}$ in. to $\frac{1}{2}$ in. or more are found between the ends of adjacent strips. Some makers have for this reason adopted the plan of putting in these packings in 2-in. lengths only. The differential expansion between the cast-iron and the brass is insufficient on these short lengths to overcome the frictional resistances to the creeping of the strip, so that the joints do not open. Where steel is used instead of brass for the casing rings of the high-pressure dummy, they can be inserted in lengths equal to the half circumference of the casing. The clearance allowed between the casing and the rotor rings depends, to some extent, on the stiffness and general dimensions of the rotor, and is commonly from 3 to 5 mils per foot of the largest diameter involved. What the true clearance is when the turbine is at work remains a little uncertain, particularly as it is quite conceivable that there is a differential expansion between the rotor and the casing which might increase clearances at one end of the dummy, and diminish them at the other. In any case it must be doubtful to the extent of the clearance allowed at the thrust block, which in the case of electric-light units is from 2 to 5 mils.

Given the clearance the amount of leakage past the dummy can be estimated as follows:—Referring to Fig. 70, the steam issues from between the first pair of rings with a considerable velocity and kinetic energy. The whole of this is destroyed by internal friction before the steam reaches the second pair of rings, which it does at a reduced pressure and augmented volume. Between these it again expands, acquiring kinetic energy, which is again destroyed before the third pair of rings is reached. This process being repeated at each pair, it will be seen that almost the whole of the available energy of the steam is destroyed by internal friction, and that it issues from between the last pair of rings with a velocity much less than that due to its complete expansion.

Subject to certain limitations, detailed below, the discharge through such a packing can be calculated by the formula

$$w = 68\,\Omega\sqrt{\frac{p_1\left(1 - \frac{1}{x^2}\right)}{V_1(N + \log_e x)}}, \qquad . \quad . \quad . \quad . \quad . \quad (45)$$

where w denotes the weight discharged in pounds per second, Ω the area available for flow in square feet, p_1 the initial absolute pressure in pounds per square inch, and V_1 the initial specific volume of the steam, whilst N denotes the number of points at which the steam is wire-drawn, and $x = \frac{p_1}{p_2}$, where p_2 denotes the absolute pressure on final discharge from the last ring of the packing.

The formula can be established as follows :—

Let the total energy available in 1 lb. of steam expanding between the initial and final pressures be U heat units. At each point of wire-drawing a certain quantity of energy must be supplied to each pound of steam to create a velocity of flow; let this be denoted by q_n heat units, which, as the steam increases in volume, will be different for each of the N pairs of rings.

The total energy expended will therefore be

$$\sum_{N=1}^{n=N} q_n,$$

which in turn must be equal to U.

Now

$$\sum_{n=1}^{n=N} q_n = \int_{\frac{1}{2}}^{N+\frac{1}{2}} q\, d\, n \text{ very approximately.}$$

It next remains to express U as a similar integral.

We may obviously write

$$U = U_N - U_0,$$

which is equal to

$$\int_{U = U_0}^{U = U_N} d\, U,$$

but this integral is taken between limits which do not correspond to those of the integral previously obtained.

It will be seen, however, that

$$U_{N+1} = U_N + q_{N+1}$$

and we may thus write approximately

$$U_N = U_{N+\frac{1}{2}} - \tfrac{1}{2}\, q_{N+\frac{1}{2}}$$
$$U_0 = U_{\frac{1}{2}} - \tfrac{1}{2}\, q_{\frac{1}{2}},$$

whence

$$U_N - U_0 = U_{N+\frac{1}{2}} - U_{\frac{1}{2}} - \tfrac{1}{2}\, \{q_{N+\frac{1}{2}} - q_{\frac{1}{2}}\}.$$

Hence, finally, the original equation may be written very approximately as

$$\int_{\frac{1}{2}}^{N+\frac{1}{2}} q\,d\,n = U_{N+\frac{1}{2}} - U_{\frac{1}{2}} - \tfrac{1}{2} q_{N+\frac{1}{2}} + \tfrac{1}{2} q_{\frac{1}{2}} = \int_{U=U_{\frac{1}{2}}}^{U=U_{N+\frac{1}{2}}} d\,U - \tfrac{1}{2}\int_{q=q_{\frac{1}{2}}}^{q=q_{N+\frac{1}{2}}} d\,q.$$

As the limits of integration correspond throughout, we may omit the integral sign and divide by q, and thus get

$$d\,n = \frac{d\,U}{q} - \frac{1}{2}\frac{d\,q}{q}$$

as the differential equation to the flow of the steam through the packing.

This gives

$$N = \int^{N} \frac{d\,U}{q} - \tfrac{1}{2}\log_{\epsilon}\frac{q_N}{q_0}$$

Now, if v be the velocity of flow through a pair of rings, we have $778\,q =$ the kinetic energy in ft. lb. of 1 lb. of the steam, or

$$q = \left(\frac{v}{224}\right)^2;$$

but if Ω be the area in square feet available for flow, V the specific volume of the steam, and w the weight passed per second,

$$v\,\Omega = w\,V,$$

whence

$$q = \frac{w^2\,V^2}{224^2\,\Omega^2}$$

and thus

$$\tfrac{1}{2}\log_{\epsilon}\frac{q_N}{q_0} = \log_{\epsilon}\rho,$$

where ρ is the ratio of expansion.

Substituting for $\dfrac{1}{q}$ in the formula for N, we get

$$N = \frac{224\,\Omega^2}{w^2}\int_{U=0}^{U=N} \frac{d\,U}{V^2} - \log_{\epsilon}\rho$$

Now it is known that when steam is wire-drawn the law of expansion approximates very closely to $p\,V =$ constant.

The work done, therefore—or, in other words, the value of U—is given by

$$U = \frac{144}{778} p_1 V_1 \log_{\epsilon}\frac{V}{V_1};$$

whence

$$d\,U = \frac{144}{778} p_1 V_1 \frac{d\,V}{V}.$$

So that

$$N = \frac{224^2 \times 144 \times \Omega^2 \times p_1 V_1}{778\, w^2} \int_{V=V_1}^{V=V_N} \frac{dV}{V^3} - \log_\epsilon \rho$$

$$= \frac{4643\, \Omega^2 p_1 V_1}{w^2} \left\{ \frac{1}{V_1^2} - \frac{1}{V_N^2} \right\} - \log_\epsilon \rho$$

Whence

$$w = 68\, \Omega \sqrt{\frac{p_1 \left(1 - \frac{1}{\rho^2}\right)}{V_1 (N + \log_\epsilon \rho)}}.$$

But in hyperbolic expansion $\rho = x$, where $x = \frac{p_0}{p_N}$ whence, finally,

$$w = 68\, \Omega \sqrt{\frac{p_1 \left(1 - \frac{1}{x^2}\right)}{V_1\ N + \log_\epsilon x}}$$

In the foregoing it has been assumed that the spaces through which the steam is wire-drawn run "full bore," but there may well be a coefficient of contraction, as defined on page 18, *ante*. In that case the discharge will be a little less than here calculated.

There is, moreover, a theoretical limitation in the applicability of the formula, just as there is to the rational formula for the flow of steam through a simple orifice. This gives incorrect results if the value of x is more than 1.73, corresponding to a limiting velocity of about 1475 ft. per second. Hence, theoretically, the above formula is only applicable if the velocity of discharge from the last pair of rings, as calculated, does not exceed this critical value. With actual packings, however, it is found that no correction is necessary, and that the discharge may safely be taken as given by the above formula, even if the limitation in question is violated.

This can be proved as follows:—Let p_1 be the pressure at the space in front of the first ring of the dummy, and p_N the pressure in the space into which the flow through the dummy is finally discharged. So long then as the pressure p_{N-1} in front of the last ring of the dummy does not exceed 1.73 times p_N, the calculated velocity of outflow will not exceed the critical value. If p_{N-1} is more than 1.73 times p_N the steam will not be fully expanded down to the latter pressure until after it has cleared the constriction.

Let w_1 be the weight discharged per second as calculated from

equation (45) above. Then for outflows in excess of the critical we have Rankine's formula . $w = \frac{p\ A}{70}$ where A denotes the area of an orifice in square inches, and from this we can find what value of p_{N-1} will give a discharge from the last pocket equal to w_1.

Thus

$$p_{N-1} = \frac{70\, w_1}{144\, \Omega}.$$

If p_{N-1} thus found is more than 1.73 times the value of p_N the critical ratio is exceeded at the last constriction. In that case assume that the value of p_{N-1} thus found is the actual value in the last pocket, and calculate from equation (45) the value w^1 of the discharge through the first (N−1) rings of the dummy.

We thus get

$$w^1 = 68\, \Omega \sqrt{\frac{p_1}{V_1} \cdot \frac{1 - \frac{1}{x^2_{N-1}}}{N - 1 + \log_e x_{N-1}}}$$

Now $\frac{1}{x^2_{N-1}}$ is always small and $\log_e x_N$ is also in general small compared with N − 1. Hence considerable variations in x_N make only very small alterations in the value of w^1. In other words w^1 will, in all such cases as are likely to occur in practice, be not materially different from w_1, so that the theoretical restriction on the applicability of the formula is actually negligible.

Packings of the type illustrated in Fig. 70 are called radial-flow packings. They have the advantage that the clearance can be adjusted by an endwise motion of the drum. They permit, however, of practically no axial play when once set, and hence where this is necessary a packing of the type shown in Fig. 71 is used. With packings of this type a greater amount of clearance is necessary, amounting to from 8 to 10 mils per foot of diameter. Fig. 71 actually represents a marine turbine gland, and, as will be seen, four Ramsbottom rings are provided at the outer end. These prevent any leakage of steam into the engine room, but will not withstand, at the speed of rotation usual, any great difference of

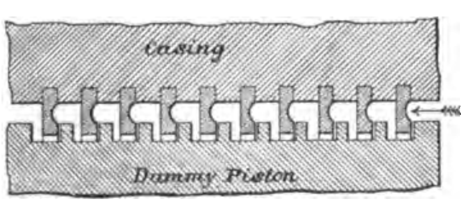


Fig. 70. Dummy Packing.

pressure. A constant leak-off is therefore allowed through the opening shown on the right, so that the total pressure taken by the Ramsbottom rings does not exceed about 2 lb. per sq. in. per ring as a maximum.

It will be seen from Fig. 71 that the fins interlace, alternate ones being fixed to the casing and to the rotor. This interlacing

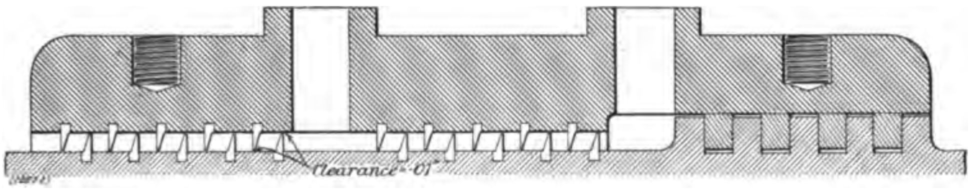


Fig. 71. Marine Turbine Gland.

is found necessary, since experiment has shown that if all the rings are fixed on the casing or on the rotor no proper wire-drawing of the steam is effected, but the latter blows straight through and the leakage loss is greatly increased.

For glands carbon packings are often used, typical types constructed by the Morgan Crucible Company being represented in Figs. 72, 73, and 74. When first introduced carbon glands often gave trouble from fracture of the segments, but by careful experiment stronger blocks have been produced, whilst at the same time more attention has been paid to the design, so as to eliminate all possibility of unfair strains on the segments. One source of difficulty at the outset arose from the fact that the shaft is naturally hotter at the internal than at the external end of the gland, and consequently expands more. Hence one large user of this type of gland has made allowance

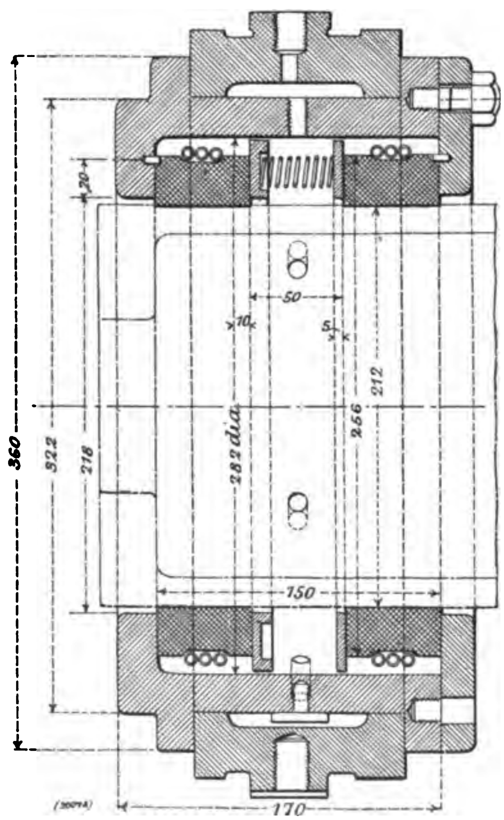
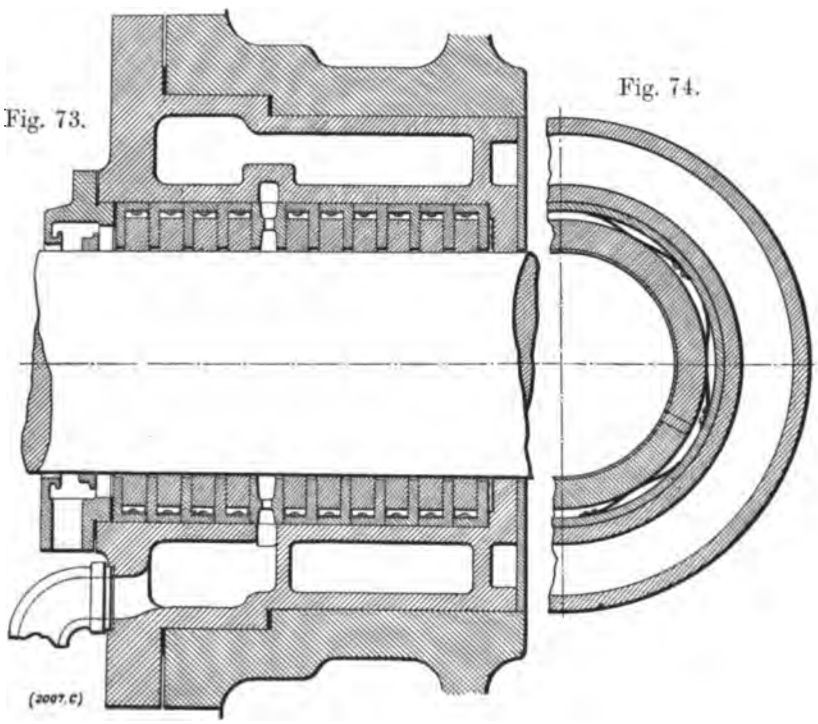


Fig. 72. Carbon Gland.

for this. At the gland the shaft is by him covered with a nickel steel sleeve on which the rings take their bearing. This sleeve is finished a few mils smaller at the inner end than at the outer. With all cold the carbon segments at the inner end are clear of the sleeve and there is some leakage. As the turbine warms up, however, this disappears, the rings coming to a bearing.

In the packing illustrated in Fig. 72 there are two carbon packing-rings, each of which is built up of three segments, which are preferably made with simple butt joints. These segments are



Figs. 73 and 74. Carbon Gland.

held together by spiral springs encircling the lot, as indicated in the engraving. Two brass rings, pressed apart by springs, as shown, keep the carbon rings up against their seats, and provision, it will be seen, is also made for introducing steam into the 'tween space. Pegs projecting from the seats on which the rings are bedded prevent these being carried round by the shaft as it rotates.

An alternative form of gland is illustrated in Figs. 73 and 74; the spiral springs holding the segments together are here replaced by flat plate springs, mounted on brass spacing rings, as shown.

Here, again, provision is made for admitting steam to the interior of the packing, thus preventing any possibility of air leakage into the turbine and thence to the condenser.

With impulse turbines of the cellular type some form of packing is generally provided to prevent leakage of steam where the shaft passes through the diaphragm.

An excellent form is represented in Fig. 75. Here the gland consists of four cast-iron segments, of which one is shown in section at A. These are serrated where they touch the shaft, and are held together by springs, as indicated at B. They are finished to a sliding fit between the plates C, C, and on these plates are shoulders, which fix a limit to the closing in of the segments. The serrations which touch the shaft are, in the first instance, finished to a diameter a few mils less than that of the shaft; and the turbine is run slowly till the serrations are rubbed away a little and the segments brought to bed on the shoulders of the parallel plates. If the shaft whips through any *contretemps*, or on passing through its critical speed, the spring, which holds the segments together, yields. These glands are practically steam-tight, and where adopted have done away with all diaphragm troubles. These were at one time very serious.

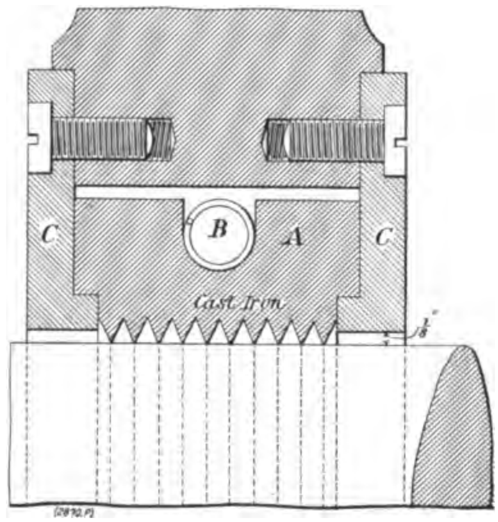


Fig. 75. Diaphragm Packing.

In the Rateau-Westinghouse machine, illustrated in a subsequent Chapter, the diaphragm packing consists simply of a bush of white metal, turned into serrations, and scraped to fit the shaft. The latter cannot seize, since the thin white metal edges readily rub away if at any point there is undue friction, and as they touch the shaft on a line only they have no tendency to distort the latter by heating.

Water-packed glands have also been used to a considerable extent. They are absolutely air-tight, but absorb a good deal of

power, and on occasions deposits of lime from the water have caused trouble. Typical examples of such glands are represented in the description of the Rateau-Westinghouse turbine, already referred to. Here the gland is interposed between the atmosphere and an enclosure, in which the pressure may reach 100 lb. per sq. in. The speed of revolution being but 750 per minute, an impeller of large diameter would be needed if reliance were placed wholly on the water gland, and to avoid this provision is also made for wire-drawing the steam through a labyrinth packing provided with "leaks-off" to parts of the turbine where the pressure is lower.

The necessity of this provision becomes immediately apparent from the fact that the power absorbed by an impeller running at a constant number of revolutions varies nearly as the fifth power of the diameter.

The water gland may be proportioned as follows:—Let D_0 be the diameter in inches of the inner circumference of the rotating ring of water on one side of the impeller, and D_0^1 the inner diameter of the rotating ring on the other side; then if p, in pound per square inch, be the pressure against which the gland is sealed we have

$$p = 0.128\left(\frac{N}{1000}\right)^2 \cdot \left[D_0 + D_0^1\right] \cdot \left[D_0 - D_0^1\right],$$

N being the number of revolutions made per minute.

If also D_1 be the extreme outer diameter of the impeller (also in inches), the horse-power absorbed by it is approximately given by the formula

$$\text{H.P.} = \tfrac{1}{4}\left\{2\left[\frac{D_1}{10}\right]^5 - \left[\frac{D_0}{10}\right]^5 - \left[\frac{D_0^1}{10}\right]^5\right\} \cdot \left(\frac{N}{1000}\right)^3$$

It thus appears that a water gland at the high-pressure end of a turbine may easily absorb something like 30 horse-power.

CHAPTER XIX.

HIGH-SPEED BEARINGS.

IN his paper on the design of high-speed rotors, published in the "Journal of the Junior Institution of Engineers," vol. xx., Mr. J. M. Newton states that it is usual to make the diameter of the journals of a turbine such that the maximum shearing stress on the metal due both to the weight and the torque is not greater than 3000 lb. per sq. in. Such a working stress is, of course, very low, and the reason for adopting it is to obtain a very stiff journal and thus ensure that its load is fairly distributed over the brass. Were the bearings made shorter it would seem that the working stress might well be increased, but designers are cautious in proceeding in this direction.

The diameter of the journal being obtained as stated, its surface speed can be deduced from the designed speed of revolution, in which case the length is fixed by the relation

$$p\,s = \mathrm{C}$$

where

$$p = \frac{\mathrm{W}}{l d}.$$

Here W denotes the total weight carried by the bearing in pounds, l the length of the journal in inches, d its diameter in inches, and s the surface speed in feet per second. The value of C in the case of turbines for generator driving varies from 2500 up to 3000. In sea practice it has commonly a value equal to about 1500.

Equivalent rules are:

$$l = \frac{\mathrm{W}}{700} \cdot \frac{\mathrm{R.P.M.}}{1000} \text{. for land turbines}$$

$$= \frac{\mathrm{W}}{350} \cdot \frac{\mathrm{R.P.M.}}{1000} \text{. for marine turbines.}$$

The work absorbed in such bearings is generally so considerable that it is necessary to cool the supply of oil.

The following rule for the heat generated per hour has been used in marine practice. Heat generated per hour

$$= d.\ l.\ s.^{1.38}.$$

This formula is also applied to the thrust block, replacing $d\ l$ by the area of the collars, and taking s as the mean speed of rubbing.

In land practice temperatures rule higher, and the viscosity of the oil being diminished accordingly the friction loss is less. From Dr. Lasche's experiments ("Traction and Transmission," vol. vi.) it appears that with a bearing temperature of 165 deg. Fahr. or thereabouts the energy dissipated in friction may be only

$$2\ d.\ l.\ s.\ \text{B.Th.U. per hour.}$$

With a bearing temperature of 90 deg. Fahr. the heat generated is about doubled. In electric-light practice the journals are some 6 mils to 8 mils slack in their bearings. If more closely adjusted than this, the loss by friction may be greatly increased (*see* Lasche, *loc. cit. supra*).

If F denote the total heat generated per hour at all the bearings belonging to a turbine, then $\frac{F}{4}$ represents the weight in pounds of oil to be supplied by the pumps per hour, and the surface needed in the cooler is equal to $\frac{F}{500}$ sq. ft. The tanks may have a capacity equal to $\frac{1}{10}$ the total quantity of oil pumped per hour.

The theoretical researches of Osborne Reynolds, and the experiments of Tower, Stribeck, and Lasche, give some reason for the belief that turbine designers have been unnecessarily liberal in fixing the proportions of their bearings. Lengthening a high-speed bearing does nothing to diminish its running temperature, since the area of oil sheared through per second is increased in the same proportion as the length, whilst the area through which heat can escape increases if anything a little less rapidly than this.

The discovery that with well lubricated high-speed bearings the journal never comes in contact with the brass was made by Mr. Beauchamp Tower, and practical experience has only served to confirm this experimental result. Turbine bearings, after a run of several years, show no signs of wear. At times a little settle-

ment due to the compression of the white metal can be detected, but there is no actual erosion.

In slow-speed bearings any heating indicates danger, and demands immediate attention, but high-speed bearings invariably run hot. Turbines in which the bearing temperature is constantly about 195 deg. Fahr. have given no trouble in practice, but a more usual limit of temperature is 165 deg. Fahr.

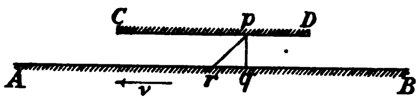
In one of Mr. Tower's experiments the head against which the oil was automatically pumped by being dragged in between the journal and its brass was no less than 625 lb. per sq. in. With slow-speed bearings, on the other hand, there is little drag on the oil; the surfaces never become more than slightly greasy. The friction is then relatively high, and the laws of friction are, at the same time, those established by Morin. Moreover, since the opposing surfaces are in direct contact, wear always occurs sooner or later.

The explanation of the pumping powers of a high-speed journal was discovered by Professor Osborne Reynolds. It lies in the fact that no well-lubricated journal is concentric with its bearing when running. If it were, the film of lubricant on it would be simply wiped off or squeezed out by the pressure carried, and this is exactly what happens with collar and pivot bearings. One gets in that case the contact of two slightly greasy surfaces, and Morin's laws of friction apply. The difference between the two cases of surfaces inclined to each other and parallel surfaces was beautifully shown by Osborne Reynolds in his paper published in the "Philosophical Transactions" in 1886. Suppose, for instance, that there are two parallel surfaces, the lower one of which, A B (Fig. 76), is moving relatively to the upper with a velocity of v feet per second. If the surfaces are merely adjacent, and there is no pressure squeezing them together, a film of lubricant between them will adhere to both, and be subjected to a shearing action. A line $p\,q$, which was vertical before the motion commenced, will be dragged out into the position $p\,r$, and there will be a certain resistance due to the viscosity of the lubricant, but the film between the plates is under no greater pressure than it is outside them.

If a load be now applied, the lubricant will simply be squeezed out, and the surfaces will ultimately come into contact.

If, however, the upper surface is inclined to the lower, as indicated in Fig. 77, then the film can sustain a pressure. Assuming this to be the case, when the steady state is attained, as much lubricant must enter across the line $P_1 Q_1$ as leaves across the line $P_2 Q_2$. The upper surface of the film is stationary, whilst the lower moves with a velocity equal to v, which is the same at both points. Since the same quantity passes each section per second, the mean velocities v_1 and v_2 must be such that

Fig. 76.



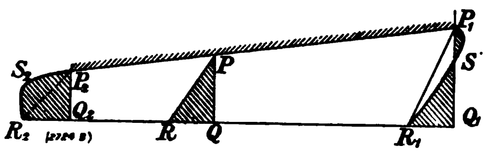


Fig. 77.

$$v_1 P_1 Q_1 = v_2 P_2 Q_2.$$

Hence the curve of velocities at the two sections is no longer represented by straight lines, such as $P_1 R_1$ and $P_2 R_2$, but at the first section it lies to the right of $P_1 R_1$, having a form similar to the curve $P_1 S_1 R_1$, whilst on the left the bulge is in the other direction, as represented by $P_2 S_2 R_2$. The hatched areas, making due allowance for the negative portion in the case of section $P_1 Q_1$, must be equal at both sections, since the volume passing is the same at both. This bulging of the velocity curve right and left can only be produced by an internal pressure, and it is this pressure which supports the load and keeps the surfaces from coming into contact. The bulging of the velocity curves takes place right and left of the line of maximum pressure, P Q say, and at this section the velocity curve is a straight line, as represented by P R, and the hatched area P R Q is equal to each of the other two hatched areas. Thus when two surfaces are inclined to one another the resistance of the lubricating fluid to shearing is transformed into an hydraulic pressure. If the flow of the lubricant becomes turbulent, then the velocity curves are no longer steady, and the condition of affairs represented cannot be maintained, and lubrication ceases. The narrower the channel way and the greater the viscosity of the fluid the higher the speed at which turbulence begins to make its appearance. Professor Osborne Reynolds not merely gave the above qualitative explanation of lubrication, but worked out mathematically from the viscosity of the lubricant the best value of the

inclination between the two opposing surfaces. In the case of journals well bedded in their brasses, this inclination is, of course, very small. From the foregoing it will be obvious that the proper place to feed oil into a high-speed bearing is along points where the oil pressure is low, and not at the top or bottom of the brasses. For examples, see subsequent Chapters.

As stated above, the lubrication of a high-speed bearing is effected by the oil being dragged in between the brasses, and this drag will be greater the greater the viscosity of the fluid. The more viscous the lubricant therefore the higher the pressure per square inch which can be carried, but at the same time the hotter the journal will run. This development of heat is, as already mentioned, not necessarily an indication of danger, in the case of a properly lubricated high-speed bearing. What is important is that the oil should be sufficiently viscous to keep the rubbing surfaces apart. Water has only about $\frac{1}{100}$ the viscosity of ordinary lubricating oil, and hence, when used as a lubricant, as, for example, in the stern tubes of destroyers, the mean bearing pressure is limited to 20 lb. to 25 lb. per sq. in. of the projected area of the bearing.

In railway-axle journals, on the other hand, a pressure of 600 lb. per sq. in. is common; grease, which is very viscous, being the lubricant.

In starting up a thoroughly lubricated journal, it is found that the friction diminishes progressively as the speed is raised, and attains its minimum value at the moment the oil first forms a complete unbroken film between the bearing surfaces. A journal run in this condition of minimum friction is, however, in a somewhat precarious state. Should the temperature rise through, say, a change in the air currents round the bearing, the viscosity of the oil will be diminished, and the drag will then be insufficient to carry the film completely round the bearing.

The pressure per square inch at which the oil film is likely to be squeezed out of the bearing depends upon the temperature of this oil film, which, it may be noted, is not necessarily that of the bearing as a whole, since heat is conducted away from the oil by the metal, and the latter is, therefore, at a lower temperature than the film. The temperature of this film is quite independent of the pressure it supports, but depends solely upon the rate at which it

is sheared, and on the rate at which heat can be conducted away through the body of the journal and brasses.

Experiment shows also that in a high-speed bearing the coefficient of friction is independent of the load.

In a paper read before the Manchester Association of Engineers, in 1907, Dr. J. S. Nicholson has given the following rational formula for the length of a high-speed bearing :

$$l = \frac{P}{40\, d^{\frac{5}{4}}\, N^{\frac{1}{4}}}$$

where P denotes the total load carried, d the diameter in inches, and N the number of revolutions per minute.

In this formula, which has not yet been fairly tested in practice, it will be seen that the greater N the shorter the bearing may be. The practice of the old millwrights was, however, to use longer bearings the higher the speed. Such bearings were, however, imperfectly lubricated, and thus liable to wear. This tendency could be diminished by increasing the bearing surfaces, and this had the additional advantage that the little lubricant supplied was less likely to be squeezed out.

The fact that great length of bearing is not essential to good running at high speeds has been established experimentally by Dr. Lasche, who successfully ran an experimental bearing $10\frac{1}{4}$ in. in diameter by $4\frac{3}{8}$ in. long under a load of $3\frac{1}{3}$ tons, or 167 lb. per sq. in., the rubbing speed being 33 ft. per second. Hitherto, however, turbine designers have not ventured far in the direction of cutting down the length of journals, though some tendency in this direction is now observable.

The self-lubricating properties of high-speed bearings are almost unique. Thrust blocks, slides, and pivots show no similar phenomenon, and so far from tending to draw oil in between the surfaces to be lubricated, their whole tendency is to extrude any existing oil film. Nevertheless, when well cooled and supplied with oil under pressure very great loads can be carried. Thus in an experiment by Dr. Lasche ("Zeitschrift Vereines Deutscher Ingenieure," 25th August, 1906), the bearing block was of gun metal, water cooled and faced with white metal. The running ring was of hardened steel, 298 mm. in outside and 185 mm. in inside diameter. Six grooves for ensuring an oil supply were cut in this ring before

hardening it. If cut in the white metal they were found to squeeze out under heavy loads.

The bearing carried for some hours a load of 15 tons when rotating at 900 revolutions a minute. This load corresponded to a pressure on the bearing surfaces of 550 lb. per sq. in., and the work absorbed in friction was 12 horse-power. Subsequent examination of the bearing showed indications of incipient fusion of the white metal, proving that this load was too high, but a load of 12 tons could apparently have been carried indefinitely. The block not having the self-lubricating properties of an ordinary cylindrical bearing had to be fed with oil under high pressure.

A self-lubricating thrust block has, however, been designed by Mr. Michell on the basis of Osborne Reynolds' researches, to which allusion has been made above, and has been successfully applied to steam turbines.

Osborne Reynolds showed that for automatic lubrication the opposing surfaces must not be parallel to one another, and in a high-speed bearing the journal is never, when running, concentric with its brass. A corresponding condition is established in the Michell thrust bearing by constructing it so that the surfaces involved adjust themselves automatically to the required inclination. This is effected by transmitting the pressure from the moving to the standing collar through a series of segmental blocks. These are cut away, as shown in Fig. 78, so that the resultant of the load P_1 at each sliding surface intersects the area involved at a point b or b_1, such that $a_1\ b_1 = 2\ b_1\ c_1$ and $a\ b = 2\ b\ c$. As a consequence the pressure is unequally distributed over the surfaces $a\ c$ and $a_1\ c_1$, and the block automatically takes a slight tilt, establishing the condition necessary for automatic lubrication. This tilt is very small, corresponding to an inclination of about 1 part in 3000.

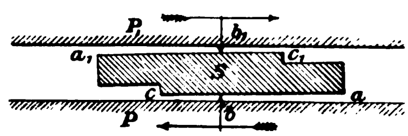


Fig. 78.
Michell Thrust Block.

The cap on the running part of this bearing is represented on the left of Fig. 79. To the right of this is shown the collar or box segments in place. One has been removed, and is shown separately below. They are held in the proper relative positions by small

set-screws as shown. The standing part of the bearing abuts on the spherical seat represented on the extreme right.

In some tests of one of these bearings a load of 800 lb. per sq. in. was successfully carried. The block was merely immersed in oil, and attended to its own lubrication. It ran quite cool, and the

Fig. 79. Components of Michell Thrust Block.

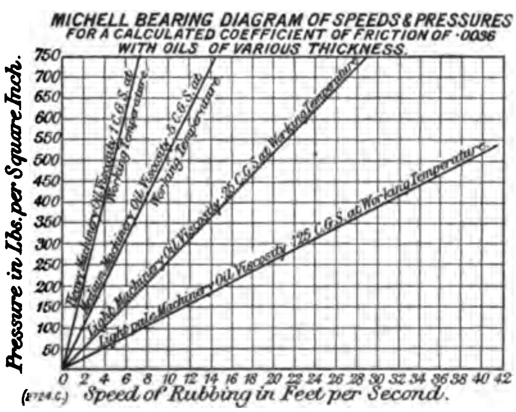


Fig. 80.

frictional resistance was too small to be recorded. A parallel test made with an ordinary collar bearing, loaded to the usual 50 lb. per sq. in., showed a coefficient of friction equal to 0.06, or practically Coulomb's value for surfaces merely greasy.

The diagram, Fig. 80, due to Mr. H. T. Newbiggin, A.M.I.C.E., shows what loads can be carried with different lubricants and different running speeds.

The coefficient of viscosity which appears in the diagram in question may, from a practical standpoint, be explained as follows: Consider a collar bearing with a single pair of collars. Let the thickness of the film of lubricant between the collars be δ cm.,

and let w be the width of the collar in centimetres, and σ its mean running speed expressed in centimetres per second. Then if W denote the work expended in friction per second expressed in gramme centimetres, the value of the coefficient of viscosity κ is

$$\kappa = \frac{W \, . \, \delta}{w \, . \, \sigma^3}$$

From the physical standpoint κ is the force required to move one square centimetre of one surface over one square centimetre of the other, at a speed of one centimetre per second, when the distance between the two surfaces is one centimetre.

In practice its value is generally determined by causing first water at its ordinary temperature to flow through a capillary tube under a certain head h. The other fluid at its working temperature is next caused to flow through the same tube under the same head. Then if the time taken for 1 cub. in. of water to pass is τ and that for the lubricant t the coefficient of viscosity for the lubricant is approximately given by the relation

$$\kappa = \frac{0.01 \, . \, \rho \, . \, t}{\tau \, .}$$

where ρ denotes the specific gravity of the lubricant at the temperature of the experiment.

CHAPTER XX.

THE STRENGTH OF ROTATING DISCS.

AS is generally known, all attempts at finding accurately the stresses which arise in a thin flat disc when it is set in rotation have uniformly resulted in failure. In practical construction, however, absolute accuracy in stress determinations is quite unnecessary, and by making certain assumptions, which are known to be fairly correct, there is no doubt but that the designer of a high-speed turbine wheel can get a very good idea as to the stresses he is imposing on his material.

Even with the approximate theory, however, the equations giving the stresses are very complicated. This is true even when the wheel is a disc of uniform thickness, and matters are aggravated greatly when this thickness is a variable one.

If, for instance, the thickness a of the disc at radius r is given by the relation $a = \frac{c}{r^\gamma}$, where c is a constant, then the approximate theory* shows that the radial stress q in such a disc (if of a substance having the density of steel) revolving at the rate of n revolutions per second is given by the relation

$$q = \mathrm{A}r^{\frac{-2+\gamma+\sqrt{4+\gamma+\gamma^2}}{2}} - \mathrm{B}r^{\frac{-2+\gamma-\sqrt{4+\gamma+\gamma^2}}{2}} + \frac{53.42\,r^2 n^2}{13\gamma - 32},$$

whilst t, the tangential stress, is given by the relation

$$t = 4.16\,r^2 n^2 + r\frac{dq}{dr} + (1-\gamma)\,q.$$

If $\gamma = 0$, we get the case of a disc of constant thickness, for which we have accordingly

$$q = \mathrm{A} - \frac{\mathrm{B}}{r^2} - 1.69\,r^2 n^2$$

---

* The equations given above were deduced by making a minimum the work done by the radial and tangential stresses, taking the latter to be functions of the radius only. Other assumptions lead to equivalent results.

and
$$t = A + \frac{B}{r^2} - 0.91\, r^2 n^2.$$

With $\gamma = 1$ the disc has a hyperbolic outline, and the stresses then depend on fractional powers of r. In the small De Laval turbines $\gamma = 2$, and here, again, the stresses depend upon fractional powers of r, a fact which makes them exceedingly troublesome to compute. The lengthiness of the calculation, moreover, increases the danger of numerical errors.

In all cases, however, the stresses on similar discs vary directly

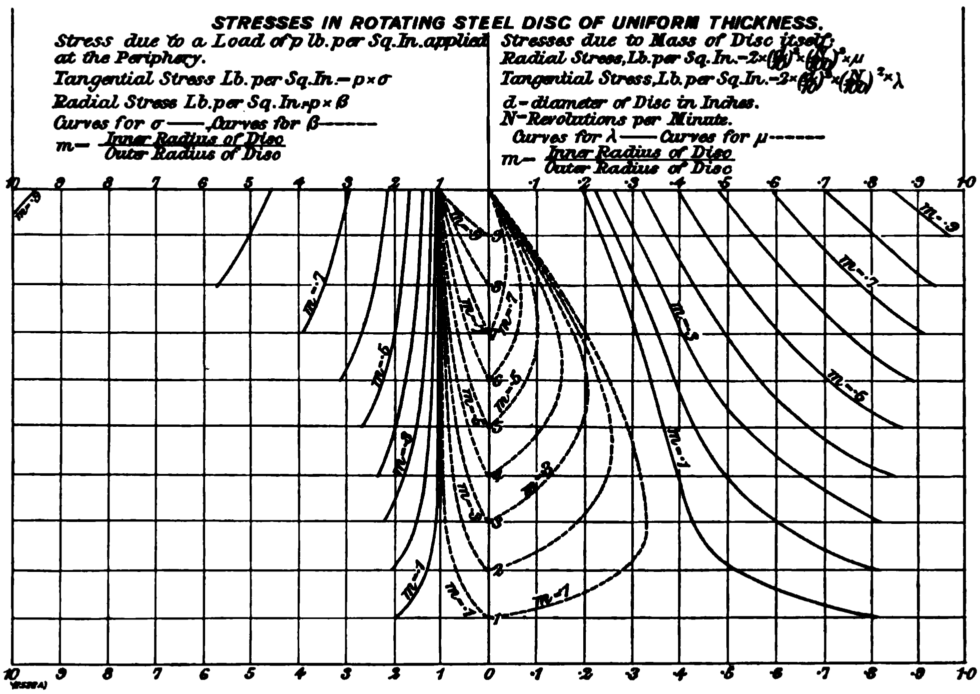


Fig. 81.

as the square of the peripheral speed. In this circumstance lies the possibility of avoiding lengthy arithmetical work by the use of curves drawn out once and for all, and in Figs. 81 to 83 curves of this kind have been plotted for three different forms of disc viz., the disc of constant thickness, the hyperbolic disc, and the disc in which the thickness varies inversely as the square of the radius.

If a thin ring of steel d in. in diameter makes N revolutions per minute, the tangential stress T developed is given by the very simple equation

$$T = 2\left(\frac{d}{10}\right)^2 \cdot \left(\frac{N}{100}\right)^2.$$

Thus a thin ring of steel 40 in. in diameter, running at 3000 revolutions per minute, would be subject to a stress of no less than 28,800 lb. per sq. in.

Now, in Fig. 81, the curves on the right show the ratios (γ or μ) which the stresses in a disc of any diameter, and of constant thickness, bear to the stresses in a thin ring of the same diameter and running at the same speed. Thus, for the case of a disc with a hole in it equal to one-tenth the external diameter, the tangential stress at

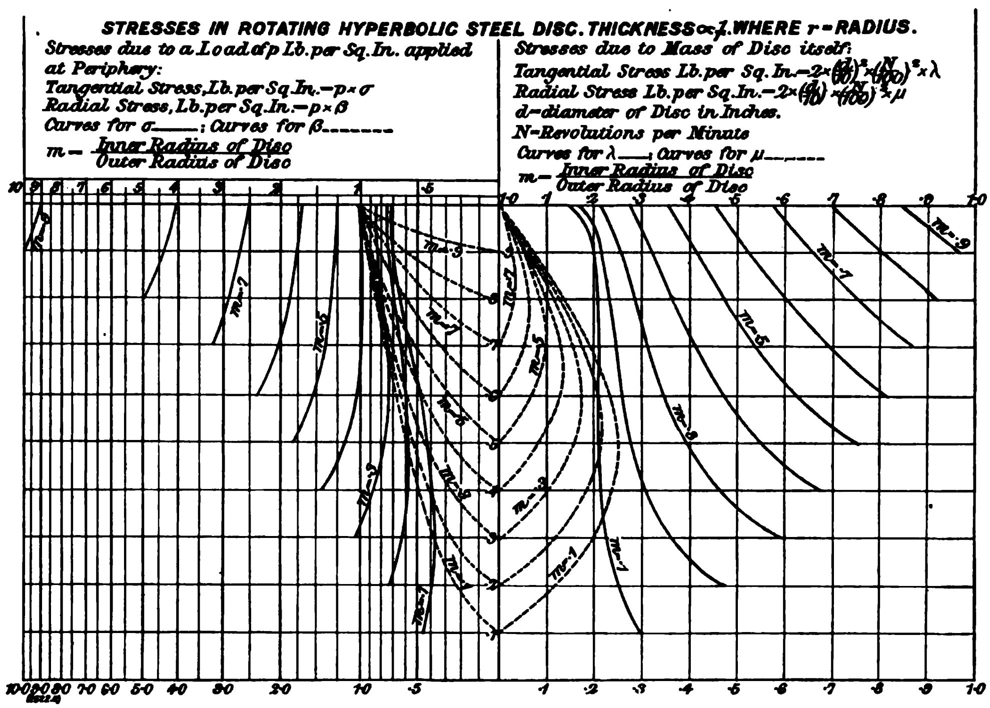


Fig. 82.

the inner diameter will be 0.815 T, or if the speed is 3000 revolutions per minute, and the diameter 40 in., the stress at the inner periphery will be $0.815 \times 28{,}800 = 23{,}470$ lb. per sq. in.

At the outer periphery of the same disc the stress will be 0.197 T, or 5670 lb. per sq. in. At intermediate points of the disc the ratio that the tangential stress bears to T is given by the height of the ordinates to the curve marked $m = 0.1$.

The ratio which the radial stresses bear to T are given by the ordinates to the dotted curve marked $m = 0.1$. This stress is zero at each periphery, and attains its maximum value (for the

case in which $m = 0.1$) at the radius $r = 0.28\ R_0$ nearly (R_0 being the external radius), and in this region it amounts to about 0.33 T, or 9504 lb. per sq. in.

If the hole in the disc is half the outer diameter, then the curves to be used are those marked $m = 0.5$. In that case, for instance, the tangential stress at the inner periphery is about 0.86 T, and at the outer about 0.4 T. The maximum radial stress occurs at a radius of about 0.7 R_0, and is equal to about 0.10 T.

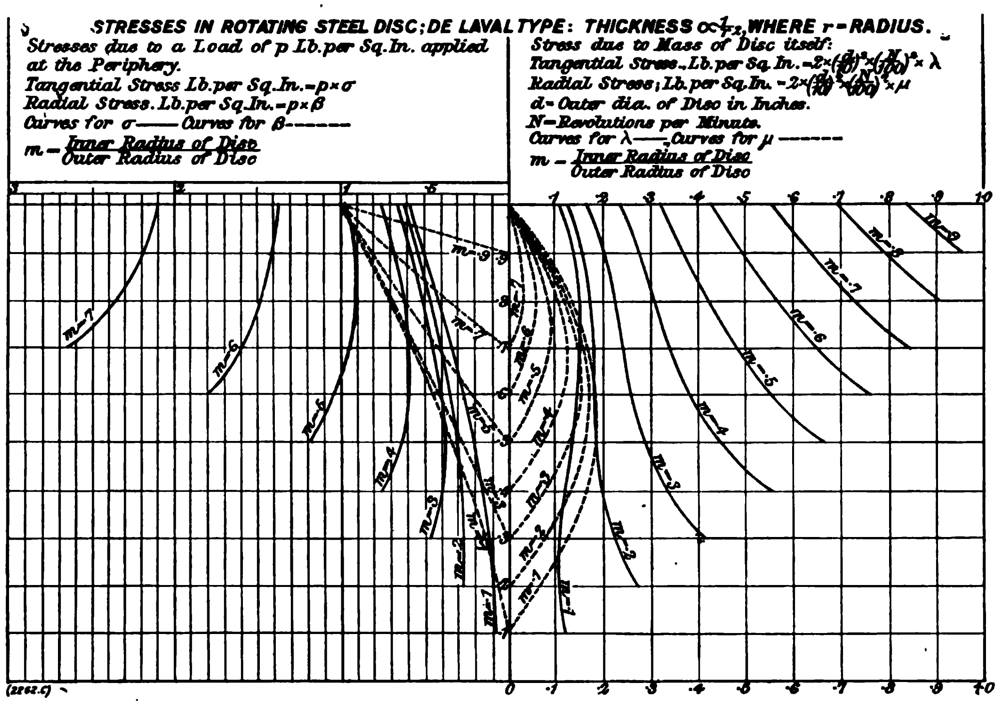


Fig. 83.

The curves on the right of Figs. 82 and 83 are exactly similar in meaning and use. In the case of a hyperbolic disc, with a hole in it equal to $\frac{1}{10}$ the outer diameter, the tangential stress at the inner periphery is, it will be seen, about 0.3 T, or less than $\frac{3}{8}$ as much as it is in the case of a disc of constant thickness.

For the case in which the thickness varies inversely as r^2, the stress at the inner periphery is still further reduced, being less than 0.12 T, and here, as will be seen, we are, with small holes, approximating to the condition of equal tangential stress throughout. With a 40-in. disc of this type, for instance, having in it a hole 8 in.

in diameter, and running at 3000 revolutions per minute, the calculated tangential stress at the inner periphery is 0.271 × 28,800 = 7800 lb. per sq. in., and at the outer periphery 0.132 × 28,800 = 3800 lb. per sq. in.

The curves on the left of each figure show the stresses caused in each disc by the application of a load of p lb. per sq. in. to the outer periphery. Thus, if a stress of 10,000 lb. per sq. in. be applied to the outer circumference of a flat disc 40 in. in diameter, having in it a hole of 8 in. diameter, the tangential stress at the inner periphery is about 2.1 × 10,000 = 21,000 lb. per sq. in., and similarly for other ratios of internal to external diameter.

The problem which generally meets a designer is to find the disc proportions necessary to support a given rim carrying one, two, or three rows of blades.

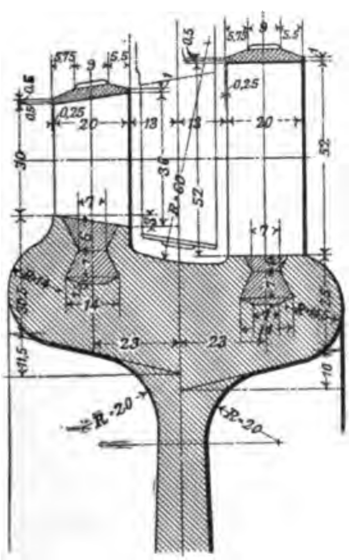


Fig. 84.

Thus, take such a rim as that represented in Fig. 84. The area of the rim, deducting that of the slots in which the blades are fitted, is 3.65 sq. in. Of this, say, 0.4 sq. in. may be taken as being merely an extension of the wheel centre, and directly supported. This leaves 3.25 sq. in. as partly supported by its own strength, and in part by the assistance it gets from the disc centre. If the mean outer diameter of the rim is 40 in., the total weight thus carried will be about 112 lb.

If the revolutions are 3000 per minute, the rim, if entirely unsupported, would be subject to a stress of about

$$2 \times \left(\frac{d}{10}\right)^2 \times \left(\frac{N}{100}\right)^2 = 28,800 \text{ lb. per sq. in.}$$

due to its own weight alone. The De Laval Company have probably had more experience with heavily strained rotating discs than anybody else, and, according to Mr. Konrad Anderssen, they permit, in the body of their wheels, stresses of about 11.2 tons per sq. in., whilst in the thinned "safety" section, just under the rim, the stress rises to over 16 tons per sq. in., which must be a

large fraction of the elastic limit, even with the high-quality steels used.

Assuming the use of a less high quality of steel, such as ordinary mild steel, we shall adopt in what follows a working stress of 16,000 lb. per sq. in. on the disc section of the wheel; but in the rim it must necessarily be less than this. In short, the rim and the disc must have the same strain, not the same stress. In the rim the axial stress is negligible, so that the increase of radius due to a tangential stress t is equal to

$$\frac{R_0 \times t}{E},$$

where E denotes Young's modulus. In the case of the disc, on the other hand, there is a heavy radial stress in addition to the tangential. If these two are nearly equal, and Poisson's ratio is taken at its theoretical value of one-quarter, the extension of the radius under a tangential stress, t_1 will be

$$\frac{R_0 \times t_1}{\frac{4E}{3}}.$$

Hence, therefore, if the working stress on the rim be taken as equal to three-quarters the working stress on the disc, or, say, 12,000 lb. per sq. in., the rim and the disc will have practically equal strains.

Hence, of the total weight of the rim $\frac{12{,}000}{28{,}800} \times 112 = 46.6$ lb., will be carried without assistance, whilst the remainder, or $112 - 46.6 = 65.4$ lb. will be transferred to the disc centre. In addition the buckets, spacing pieces and shrouding will add a weight of, say, 67 lb. more. Hence the total weight, W, carried by the disc will be 132.4 lb. When the wheel is in rotation, this weight will develop centrifugal forces, the intensity of which, reckoned in pounds per inch run of the disc circumference, is given by the relation

$$\text{Load in pounds per inch run} = \frac{W}{22} \cdot \left(\frac{N}{100}\right)^2 = 5410.$$

If we allow 16,000 lb. per sq. in. as the working stress, the thickness of the disc at the edge will be 0.338 in.

Suppose the effective inner diameter is 6 in., then a reference to Fig. 81 shows that if a disc of constant thickness be used, the tangential stress at the inner periphery, due to the above edge-

loading, will be about $2.1 \times 16{,}000 = 33{,}600$ lb. per sq. in., so that the use of such a disc is out of the question.

Next assume the disc to be of the hyperbolic type. From Fig. 82 it appears that the tangential stress at the inner periphery, due to a radial stress of 16,000 lb. per sq. in. at the outer, amounts to about $0.65 \times 16{,}000 = 10{,}400$ lb. per sq. in.

At the same time, however, a reference to the right-hand side of the same figure shows that the tangential stress due to the mass of the disc itself amounts at the inner periphery to about 0.4 T, where

$$T = 2 \times \left(\frac{d}{10}\right)^2 \times \left(\frac{N}{100}\right) = 28{,}800 \text{ lb. per sq. in.}$$

Hence $0.4\,T = 11{,}520$, so that the total stress at the inner circumference will be $11{,}520 + 10{,}400 = 21{,}920$, which is too great. Thirdly, let the disc be of the third type. Here the tangential stress at the inner periphery due to the radial load is, from Fig. 83, about $0.17 \times 16{,}000 = 2720$ lb. per sq. in.; and the stress due to the mass of the disc itself will be $28{,}800 \times 0.19 = 5470$, making a total of 8190 lb. per sq. in. Hence the factor of safety is more than ample, but the thickness of the wheel at the inner periphery is very great, being $44\frac{1}{2}$ times the thickness at the rim, or 15 in. We may, however, by interpolation find the exact value of γ which will give a maximum stress of 16,000 lb. per sq. in. at the inner periphery.

Let the thickness of the wheel be denoted by

$$a = \frac{C}{D^{\gamma}}.$$

We can tabulate the following stresses for different values of γ:—

γ.	Stress Due to Rim Load.	Stress Due to Mass of Disc Itself.	Total Stress.
0	33,600	23,600	57,200
1	10,400	11,520	21,920
2	2,720	5,470	8,190

If we plot these, as in Fig. 85, we find that the required stress at the inner circumference will be attained if

$$a = \frac{C}{D^{1.30}}.$$

In this equation the constant C can be calculated once the rim

thickness is known, and this thickness, as calculated on page 190, was 0.338 in. Thus, when D = 40 we have a = 0.338 in., as found above, whence C = 40.89. Using this value of C we find that when D = 6, a = 3.981 in.; when D = 20, a = 0.8328 in.; and when D = 30, a = 0.401 in. Having now these dimensions, a profile of the disc can be sketched in.

So far as the stresses at the inner periphery are concerned, a disc having the above dimensions would suffice. An examination of Figs. 81, 82, and 83 shows, however, a possible danger point, at diameter D = 0.9 × 40 = 36 in. Tabulating values for this region we get the following table:—

γ.	Tangential Stress Due to a Radial Stress of 16,000 lb. at Rim.	Tangential Stress Due to Disc's Own Mass.	Total Tangential Stress.
	lb. per square inch.	lb. per square inch.	lb. per square inch.
0	16,800	7200	24,000
1	14,400	6050	20,450
2	9,975	4180	14,155

Plotting these values in Fig. 85, it appears that the stress at 36-in. radius, taking γ as 1.3, is about 18,800 lb. per sq. in. We can reduce it either by making the disc thicker throughout, or by increasing the value of γ to, say, 1.75. If we adopt the former course, and maintain the value of γ as 1.30, we find by plotting that the 18,800 lb. per sq. in. total stress, found above to exist at 36 in. in diameter, is made up of two elements—viz., 13,300 lb., due to the edge-loading, and about 5500 lb. as due to the mass of the disc itself. The latter will be unaltered if we increase throughout the disc thickness in the same proportion, whilst the former stress is inversely proportional to the disc thickness. Since the total stress is not to exceed 16,000 lb., the stress due to the edge-loading must not exceed 16,000 − 5500 = 10,500. Hence the thickness of the disc must be increased throughout in the ratio of $\frac{13,300}{10,500}$. We thus get thickness at 40 in.

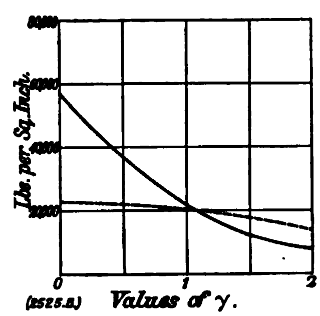


Fig. 85.

= 0.428; at 30 in. = 0.621; at 20 in. = 1.056; at 6 in. = 5.18 in.

Having obtained a theoretical disc profile as above, a practical form may be drawn to include it, as indicated in Figs. 86 and 87, where the theoretical profiles are represented by the dotted lines, and outside them a practical form by the solid lines.

THEORETICAL AND PRACTICAL PROFILES.

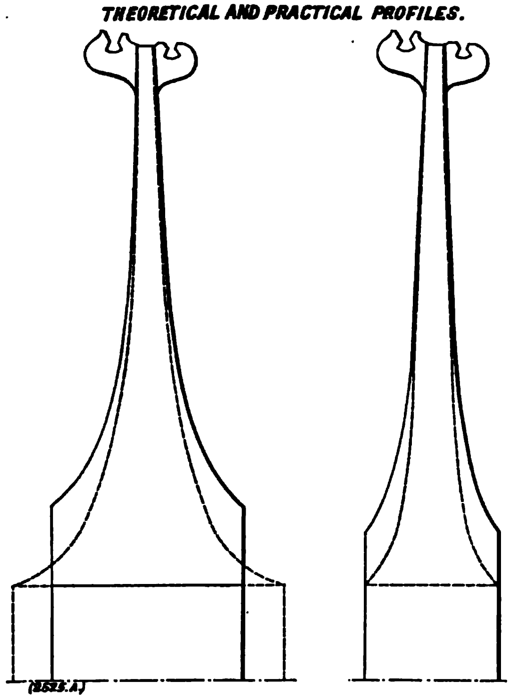


Fig. 86. Fig. 87.

Obviously the thin portions of the theoretic profile near the shaft can be of very little assistance in practice, however effective they may be on the approximate theory necessarily adopted as a basis of calculation. Hence a practical profile, such as that shown, must be put in by the exercise of the designer's judgment. The necessity for this procedure makes it absurd to use great refinements in the calculation of the strength of such wheels. The extra metal added, being near the hub, adds little, it may be noted, to the total weight of the wheel.

Turbine wheels must be made a tight fit on their seats, so as to avoid any risk of them getting loose when in work. This, of course, implies that the inner periphery of the wheel is strained when the wheel is forced into place, and the question arises as to how far it is necessary to take such straining actions into account. The answer given is largely a matter of individual judgment In the writer's view provision for seat stress is unnecessary, for several reasons. In the first place, stresses due to deformations are materially different in character from those due to applied loads. A load "is always at it," but an excessive stress caused by a surface deformation is relieved by the plastic yield of the material. Thus it is not easy to burst a mild-steel ring by drifting it out; whereas were an internal pressure applied giving the same

calculated stresses, the ring would fly in pieces. Further, when a wheel is run up to speed, the inner periphery expands, under the stresses induced by the centrifugal forces, thus relieving to a large extent those due to the forcing of the wheel on to its seat. Finally, the necessity for adopting a practical profile greatly strengthens the hub of the wheel.

Those who hold a contrary view as to the importance of stresses arising from forced fits can, however, deduce the stresses arising from loads applied to the inner periphery of a wheel by using the same curves as are employed in finding the stresses due to loads applied at the outer periphery. Thus, if the stress at any radius due to a stress p applied at the outer periphery is σp, then, if the stress p be applied instead to the inner periphery, the corresponding stress at the same radius will be $(1 - \sigma) p$.

The curves given apply solely to wheels having central holes. Where solid wheels are used, the most economical section is that of Dr. de Laval. If p be the radial stress permitted at the outer periphery of the disc of diameter D, then t the thickness of the disc at this point can be calculated. The thickness at any other diameter d of the solid disc is then given by the relation

$$a = t \,.\, \epsilon^{\frac{1}{p}\left(\frac{N}{100}\right)^2 \left[\left(\frac{D_1}{10}\right)^2 - \left(\frac{d}{10}\right)^2\right]}.$$

Here D and d are both taken in inches, and $\epsilon = 2.718$.

The stress throughout the disc has then everywhere the same value p.

CHAPTER XXI.

GEARED TURBINES.

DR. DE LAVAL was the first to realise the possibilities of geared steam turbines. By the use of high-speed gearing it is possible to drive machinery at a moderate rate of rotation by means of a turbine turning with a very great angular velocity. Thus in the case of small Laval turbines a speed of 30,000 revolutions at the wheel is reduced by gearing to 3000 revolutions. Within certain limits, the higher the turbine speed the more cheaply is it possible to attain a given standard of efficiency. It has, however, to be observed that gearing is not universally applicable. Very large turbines must always run at a moderate angular velocity. It has been found quite possible to build turbines of 300 to 500 horse-power to run at 10,000 revolutions per minute, but it would not be possible to construct an efficient 3000 horse-power turbine to run at the same speed. In fact, for a given heat drop the blade speed should be the same for the large as for the small turbine, that is to say, the mean diameter of the buckets should be the same. To get ten times the power from the wheel the buckets should therefore be ten times as long, and would thus become practically mere spokes fitted to a central boss.

Direct connection will therefore continue to be the rule in the case of turbines of any considerable size driving electric generators; but for small outputs the geared turbine has a substantial advantage, and for special purposes, such as propeller driving, gears transmitting some thousands of horse-power, may, with advantage, be used to connect up a slow-speed propeller to a high-speed turbine.

In the De Laval turbines the pitch line speed of the gear is as high as 100 ft. per second. The teeth are short and numerous, as indicated by the table on the next page, showing the pinion sizes used in different cases.

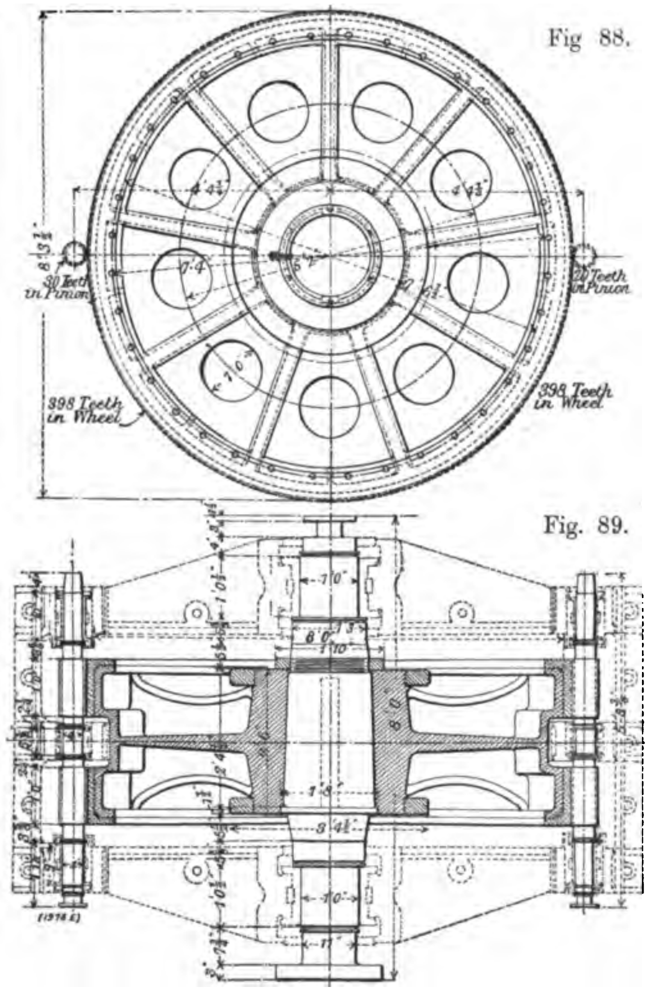
The tooth pressure allowed per lineal inch is about 10 lb. in the smallest sizes, 30 lb. in a 30 horse-power gear, and 45 lb. in the gears for a 300 horse-power turbine. These are much less than is permitted in motor-car practice, where, with wheels of chrome-vanadium steel, case hardened, tooth pressures of 1300 lb. to 1500 lb. per lineal inch have been freely adopted, with pinions having about five teeth per inch of diameter and a pitch line speed of about 15 ft. per second. These motor-car gears have, of course, straight teeth, whilst double helical gearing is invariably used for turbines.

H.-P. of Turbine.	Outside Diameter of Pinion in Inches.	Number of Teeth.	Depth of Teeth in Inches.
10	1.077	21	0.075
75	1.53	19	0.1169
110	1.82	23	0.1169
300	2.65	31	0.1275

The first large high-speed gear to be used in service was that installed in the "Vespasian" by the Parsons Marine Steam Turbine Company in 1910. A view of this gear is given in Figs. 88 and 89. The gear wheel is 8 ft. $3\frac{1}{2}$ in. pitch diameter, and has 398 teeth with a circular pitch of 0.7854 in. The material is forged mild steel. The teeth are inclined at an angle of 20 deg. to the axis of the wheel. The pinions are of chrome-nickel steel having a tensile strength of 37 tons and an elastic limit of 32 tons per sq. in. Each pinion is 5 in. in diameter having 20 teeth, so that the gear ratio is 19.9 to 1. As shown, there are two pinions on opposite sides of the main gear wheel. One of these pinions is driven by the high-pressure turbine and the other by the low-pressure. The pitch-line speed is about 30 ft. per second, and the pressure on the teeth about 370 lb. per lineal inch. The efficiency of the gear is about $98\frac{1}{2}$ per cent. The wear after two years' running was found not to exceed 2 mils, and even this is attributed to the fact that settling chambers were not in the first instance provided for the oil used in lubricating the gears, so that the latter carried up a little grit into the teeth. On subsequently making provision to keep the oil clean, the teeth took and kept a fine polish.

In another installation fitted at Calderbank steel works to drive

a rolling mill, the turbine, designed to develop 750 brake horse-power, runs at 2000 revolutions per minute. The high-speed pinion is 7.143 in. pitch diameter and has 25 teeth, the gear wheel having 131 teeth, and the gear ratio is, therefore, about 5.2 to 1. The pitch-line speed is 62.4 ft. per second, and the tooth pressure per



Figs. 88 and 89. The Gearing of the "Vespasian."

lineal inch is 275 lb. The gears as before are double helical, the teeth being hobbed to spirals set at 23 deg. to the axis of the wheels.

In America some very large high-speed gear wheels have been constructed by Mr. George Westinghouse. One of these, designed

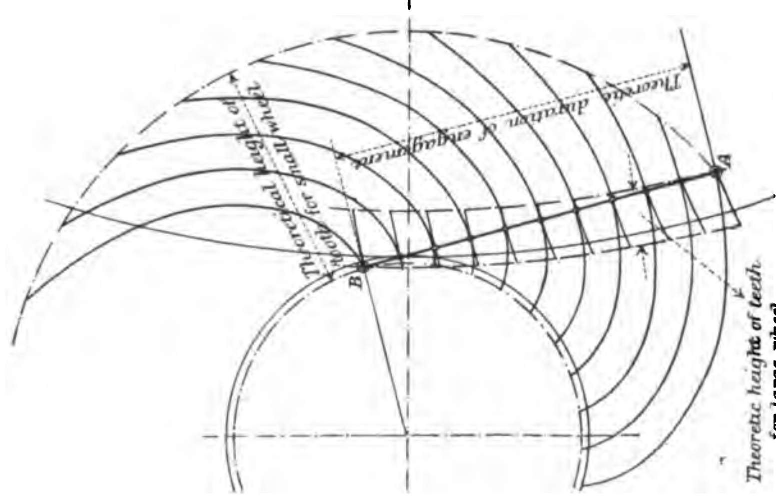


Fig. 90.

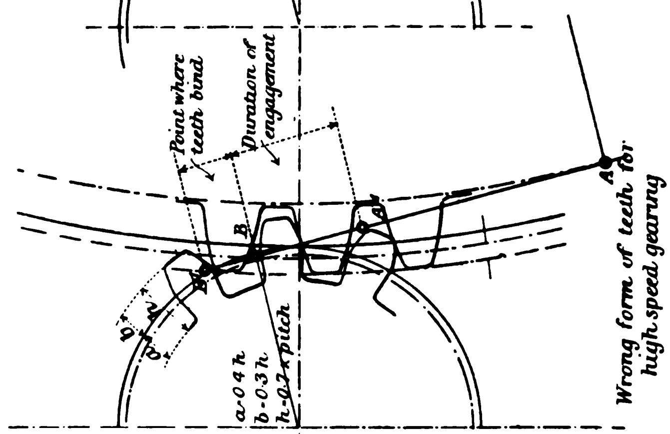


Fig. 91.

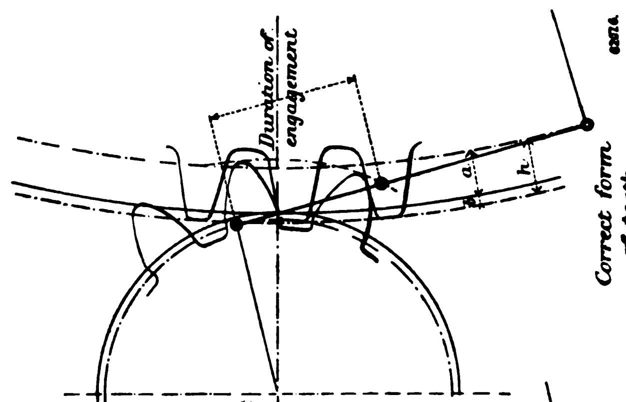


Fig. 92.

to transmit 6000 horse-power with a gear ratio of 5 to 1, had pinions with thirty-five teeth and $1\frac{1}{4}$ in. circular pitch.

The helices in this case make angles of 30 deg. with the axis of the shaft. The turbine runs at 1500 revolutions per minute, so that the pitch line speed is about 92 ft. per second. The normal pressure on the teeth per lineal inch is 1180 lb. Careful tests (see ENGINEERING, Dec. 3, 1909) showed an efficiency ranging between 97.8 to 99 per cent. In this case the pinion is supported in a "floating frame," a device originally introduced by Messrs. Melville and Macalpine, and intended to allow the pinion to adjust its angular position with regard to the main wheel.

The question as to whether or no the fitting of such an arrangement is advisable must be settled by practical experience, but there are certain theoretical considerations which for the present (1912) make its utility somewhat doubtful.

In view of the very high speed of the pitch line and the considerable inertia of the pinion, it does not seem possible that the latter can, in practice, adjust itself to provide for faults in the teeth. Again, if any such adjustment does take place the pinion will be tilted axially, and its teeth "set across" those of the gear wheel. There is thus much reason to believe that the success of the system depends mainly on its never being called upon to act.

In gearing down turbines a considerable reduction ratio is generally required, and the teeth should be designed accordingly. The following note on this subject is taken from "Traction and Transmission," vol. vii., page 122.

"Generally speaking, a small pinion does not gear well with a large wheel, at least with teeth of the involute type. To avoid interference it is accordingly a common practice to ease off the points of the teeth, which then, of course, are no longer of the true involute form. The nature of this interference is well shown in Fig. 91, in which the teeth are made in the usual way, the point being 0.3 pitch and the root 0.4 pitch, making the total height of the teeth 0.7 pitch. It will be seen that the points of the teeth A^1 and B^1 are liable to jam. On plotting successive positions of the interacting tooth faces, however, as in Fig. 90, it will be found that whilst the point of contact between two teeth sweeps over the whole face of the pinion tooth, only a small fraction of the face of the wheel

tooth is utilised, and hence the theoretical height of the pinion tooth is, as Fig. 90 shows, much greater than the theoretical height of the wheel-tooth gearing with it. By taking advantage of this fact involute gears can be designed, in which a small pinion will gear perfectly well with a large wheel without the slightest danger of interference. A set of teeth designed on these lines is shown in Fig. 92. It will be seen that the pinion teeth extend far above the pitch line, and are almost all addendums, whilst the wheel teeth are practically all roots. The total height of the teeth is made from 1.8 to 2 the diametral pitch, the latter being taken as circular pitch divided by π."

CHAPTER XXII.

THE CONDENSER.

THE turbine is the only form of steam-operated prime mover which is able to derive a fair degree of advantage from really high vacua, but even the turbine is incapable, when constructed as a commercial machine, of benefiting as much by extremely low back pressures as it theoretically should. Thus Mr. M. G. I. Swallow (Transactions of the North-East Coast Institution of Engineers, 1911) has made the following comparison between the saving theoretically due to different increases in vacua and the figures actually realised with a large turbine designed to work with a vacuum of 28¼ in. :—

Increase of Vacuum.	Theoretical Saving in Steam per Cent.	Actual Saving in Steam per Cent.
27 in. to 28 in.	6.8	5
28 in. to 29 in.	11.4	6
29 in. to 29½ in.	16.0	6.6

The fact remains, however, that although at high vacua the economy to be realised is but a fraction of what is theoretically due, the gain is still substantial, and it is therefore very desirable to supply steam turbines with condensers of the highest efficiency. In fact, the introduction of the turbine has led to great improvements in the design of the condenser, the pioneering work in this direction being mainly due to Sir Charles A. Parsons.

The theoretical limit to the vacuum possible in a condenser is, of course, fixed by the temperature of the circulating water. The diagram (Fig. 93, page 202), reduced from one contributed to the Journal of the Junior Institution of Engineers in 1912, by Mr. J. M. Newton, B. Sc., shows the vacuum theoretically possible with cooling water

supplied at the different temperatures marked on the curves, and with different ratios of Q, the weight of circulating water supplied, to W, the weight of steam condensed. It will be noted that to obtain a high vacuum a low temperature of cooling water is much more important than the quantity supplied. With cooling water at 80 deg. Fahr., supplied in the ratio of 70 lb. to 1 lb. of steam condensed, the theoretical vacuum is just over 28.3 in.; whilst if the cooling-water temperature is 40 deg., the theoretical vacuum is raised to nearly 29.6 in. It is further noteworthy that when the ratio of cooling water to steam is increased beyond 70 to 1 the vacuum rises very slowly, and the power expended in pumping may then exceed the gain derived. In power-station practice under average conditions the ratio $\frac{Q}{W}$ is commonly between 60 and 70. The temperature rise of the cooling water may be taken as equal to $\frac{1050\,W}{Q}$.

The curves in Fig. 93 show the vacuum theoretically due, but in practice this is for a number of reasons never realised. The most important of these disabilities is to be found in the presence of air in the condenser, part of which obtains access from leaky joints, whilst part is carried into the boiler in solution in the feed water.

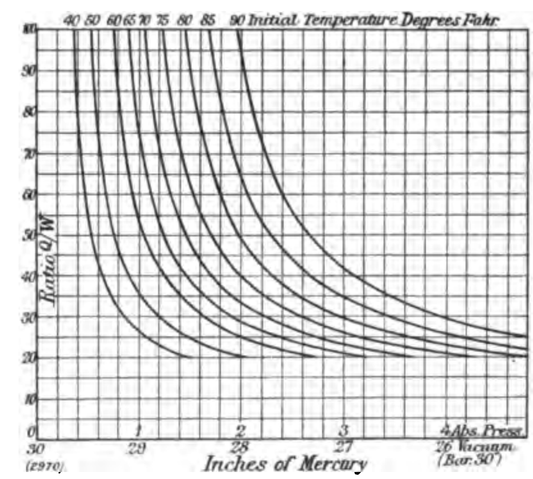


Fig. 93.

Mr. D. B. Morison has made many experiments on this latter point. He finds (Transactions of the Institution of Naval Architects, 1908) that the fresh water carried in ships' tanks commonly holds in solution 2 to $3\frac{1}{2}$ volumes of air for every 100 volumes of water. Condensed water as drawn off from an ordinary condenser, through the air pump, contains $1\frac{1}{2}$ to 2 volumes of air (measured at normal pressure) per 100 volumes of water; but if withdrawn from the upper part of the condenser, without being allowed to shower

down through the air-logged lower portion, it may, he states, contain practically no air. If the feed is pumped into a feed tank without disturbance, and fed into the boiler by a float-controlled feed pump, no further aeration of it occurs. Mr. Morison's figure of $1\frac{1}{2}$ to 2 volumes of air per 100 of feed water is equivalent to about $2\frac{1}{4}$ lb. of air per 10,000 lb. of feed. Mr. Newton, in the paper already referred to, states that in good commercial power-station practice, the air pumps ought not to be called upon to remove more than 4 lb. of air per 10,000 lb. of steam condensed. By taking special care this figure can, in fact, be very substantially reduced; and Professor Josse has, in the case of a 300-kw. Parsons turbine, succeeded in reducing the ratio in question to rather under 1 lb. of air per 10,000 lb. of water evaporated. Every pound of air which enters the condenser has to be removed by the air pump, and at high vacua the volume occupied by 1 lb. of air becomes enormous.

If the temperature of the interior of a condenser, near the steam inlet, be measured with *precision* (a matter of some difficulty), it is found that this temperature is that corresponding to the pressure as given by an ordinary steam table. Thus, if the temperature here observed be 101.4 deg. Fahr., a reference to a steam table shows that the corresponding pressure is 0.980 lb. absolute, or 2 in. of mercury, and is equivalent accordingly to a vacuum of 28 in., which is, in fact, the vacuum which would be registered by an accurate gauge attached to the condenser at the top. A similar gauge fixed on the air-pump suction would give precisely the same reading, there being no appreciable difference in pressure between the top and bottom of an ordinary condenser. A thermometer reading taken at this point is however invariably lower than that registered at the steam inlet, and this difference is greater the greater the proportion of air to steam which enters the condenser. Two such thermometers may therefore be employed as a sort of air gauge. The closer their readings, with a stated vacuum, the less is the air entering the condenser.

At the air-pump suction, in fact, the total pressure registered is the sum of the "partial" pressures of the steam and of the air there present. The pressure at this point exerted by the steam is obtained from the temperature on referring to a steam table.

Thus, if the thermometer reading here were 80 deg. Fahr., the steam table shows that the pressure exerted by the steam in this locality is 1.02 in. of mercury. The temperature at the steam inlet was, however, 101.4, corresponding to an absolute total pressure of 2 in. of mercury. The difference must be due to the air, which accordingly exerts a pressure of $2.00 - 1.02 = 0.98$ in. of mercury. One pound of

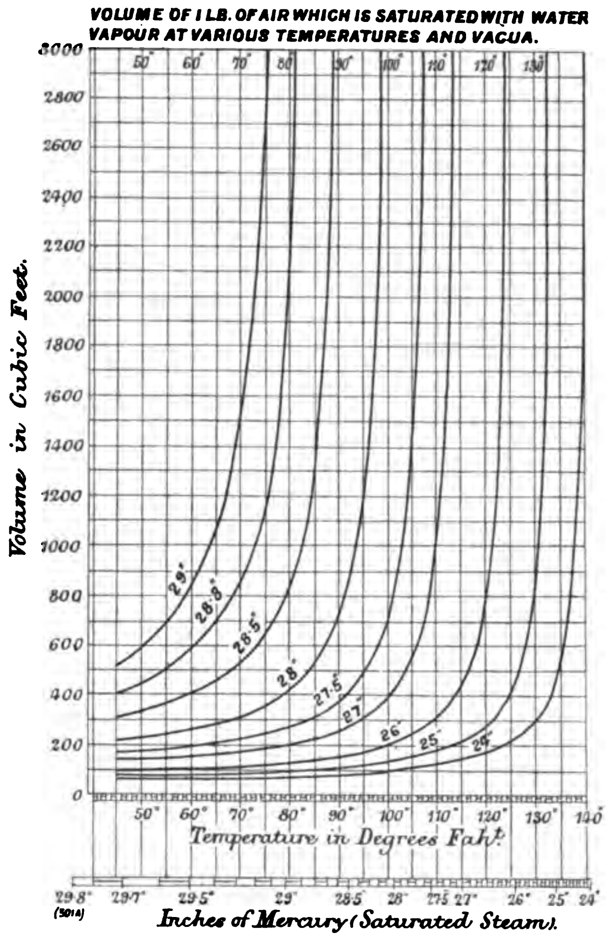


Fig. 94.

air at the atmospheric pressure of 30 in. and an absolute temperature of 518.4 deg. Fahr. occupies a volume of 13 cub. ft. Hence at a pressure of 0.98 in. of mercury and an absolute temperature of 459.4 deg. + 80 deg., the volume occupied will be

$$V = 13 \times \frac{539.4}{518.4} \times \frac{30}{0.98} = 345 \text{ cub. ft. per lb.}$$

Hence for each pound of air entering per minute the air pump must have an effective displacement equal to the above figure.

The necessity of such a calculation as the above may be avoided by using the curves given in Fig. 94, which are taken from a paper read by Mr. D. B. Morison before the Institution of Naval Architects in 1908.

If, now, the quantity of air which enters the above condenser is doubled, the air-pump capacity and the flow of circulating water being the same as before, the effect on the vacuum can be readily realised. In order to get away the doubled quantity of air, the air pump, having a fixed displacement, must receive it at twice the pressure. Hence the partial pressure of the air rises to $2 \times 0.98 = 1.96$ in. of mercury. Adding the partial pressure of the steam, which is 1.02 in. of mercury at 80 deg. Fahr., the temperature of the air-pump suction, the total pressure inside the condenser becomes $1.96 + 1.02 = 2.98$, corresponding to a vacuum of only just over 27 in. This rise would reduce the efficiency of a turbine by about 5 per cent.

The results of a very instructive set of comparison tests of a 300-kw. plant, showing the detrimental effect of air, have been given by Mr. Newton in the paper already quoted, and are reproduced below:—

Test Number.	1	2
Weight of steam condensed per hour, lb.	6,500	6,600
Cooling water, quantity per minute, gallons	500	330
,, temperature at inlet, deg. Fahr.	52	48
,, ,, at outlet ,, ...	73	80
Temperature of steam entering condenser, deg. Fahr. ...	102	84
Temperature observed at air-pump suction ,, ...	65	77
Corresponding partial pressure of air, lb. per sq. in. ...	0.70	0.11
Weight of air entering condenser per hour, lb.	20	2.1
Power taken by condensing plant, kilowatt	13.8	6.6

The turbine in question was rated at 300 kw., and the first set of tests were made immediately after erection. The air leakage was excessive, making necessary an abnormal expenditure of power in running the pumping plant. An examination of the copper bellows pipe between the turbine and the condenser disclosed a

number of small pin holes, and on stopping these the results obtained on the second trial were realised. It will be seen that the quantity of cooling water has been reduced by one-third, but in spite of this the vacuum has been increased by three-quarters of an inch, which reduced the steam rate of the turbine by $4\frac{1}{2}$ per cent. At the same time the feed temperature has been raised, and the power taken to operate the condenser halved. The net result was an improvement of about 7 per cent. in the over-all economy of the plant.

Mr. Newton remarks that the leaks in this case were much too small to be detected by means of a candle flame, and recommends that in testing a condenser for leakage the air-pump discharge should be blanked off and the condenser, connecting pipe, and turbine filled with water up to the level of the spindle glands. In this case water will drip from every leak, however minute, and the place being marked with chalk, the hole can be stopped at the end of the test.

It is quite as important to make certain that the joints are tight, between the condenser and such fittings as gauges and the like, as that the main joints have been properly made. An opening equivalent to a hole $\frac{1}{32}$ in. in diameter will pass air into a condenser at the rate of, roughly, 1 lb. per hour, which is quite enough to lower appreciably the over-all efficiency of the plant.

The proper method of dealing with such air leakage is to find out and stop the leaks, but other alternatives are possible. The most obvious is to increase the effective displacement of the air pump, and this has frequently been resorted to. Again, the supply of circulating water may be increased, thus lowering the temperature of the air-pump suction, and with it the partial pressure of both the steam and the air. This plan is, however, only feasible if the quantity of air which leaks in is only moderately in excess of the normal, and, if adopted, the air pump should be of the "dry" type, so as to avoid lowering the temperature of the condensate at the same time as the air temperature is lowered.

Air in a condenser has a further detrimental effect in addition to its action as above set forth. It is, in fact, an excellent non-conductor, and as such "blankets" the tubes, preventing

a free interchange of heat between the steam and the circulating water.

Such experiments as have been made on the importance of this effect have not been wholly concordant. Osborne Reynolds, who appears to have been the first to appreciate the important part played by air in checking condensation, found that a very small addition of air had a very pronounced effect, but that subsequent additions were not proportionately detrimental. Experiments made by Mr. J. A. Smith, and described in ENGINEERING, March 23, 1906, indicated that the effect of small additions of air was not as serious as would appear from the experiments of Osborne Reynolds, but there is, perhaps, some doubt as to whether in any of Mr. Smith's experiments the steam was quite air free, since he refers to the steam as "pure, or *nearly* pure." Callendar ("Enc. Brit.," article: "Heat") says that with air-free steam the difference in temperature between the steam and the wall on which it condensed was 36 deg. Fahr. for a condensation of 109 lb. of steam per sq. ft. per hour, which is equivalent to a temperature difference of 0.33 deg. Fahr. for each pound condensed per square foot per hour. In actual surface-condenser practice the difference in temperature between the steam and the outer surfaces of the tubes is certainly very much more. It should be noted in this connection that the average temperature of the outer surface of the tubes is often lower than that of the condensate, since the cold water dropping from the tubes condenses further steam as it falls, and has therefore its temperature raised.

The fall of temperature in the wall of the tube itself is (in the case of $\frac{3}{4}$-tubes of No. 18 gauge) about $\frac{1}{12}$ deg. Fahr. for each pound of steam condensed per square foot per hour.

In power-station practice from 8 lb. to 9 lb. of steam are commonly condensed per square foot per hour, whilst the supply of water is from 60 lb. to 70 lb. per pound of steam condensed. The velocity of flow in the tubes is generally between 3 ft. and 4 ft. per second. Poor results will, however, be realised unless sufficient air-pump capacity is provided to keep down the partial pressure of the air, and if the air leakage is large no reciprocating pump can be expected to deal effectively with the volume to be removed. This can, however, be accomplished by the "vacuum augmenter," invented

by Sir C. A. Parsons, which is illustrated in Fig. 95. Here the air-pump inlet is placed about 4 ft. below the lowest point of the condenser, to which it is connected by a bent pipe, as shown. As a consequence, the pressure at the air pump is greater than that in the condenser by about 2 in. of mercury. Hence the same weight of air can be removed by a much smaller pump. The condensed steam enters the air pump by the bent pipe, but the air is drawn off by a steam jet situated as shown. This discharges into an auxiliary condenser, which has about one-twentieth of the cooling surface of the main condenser, and this drains to the air-pump inlet, as

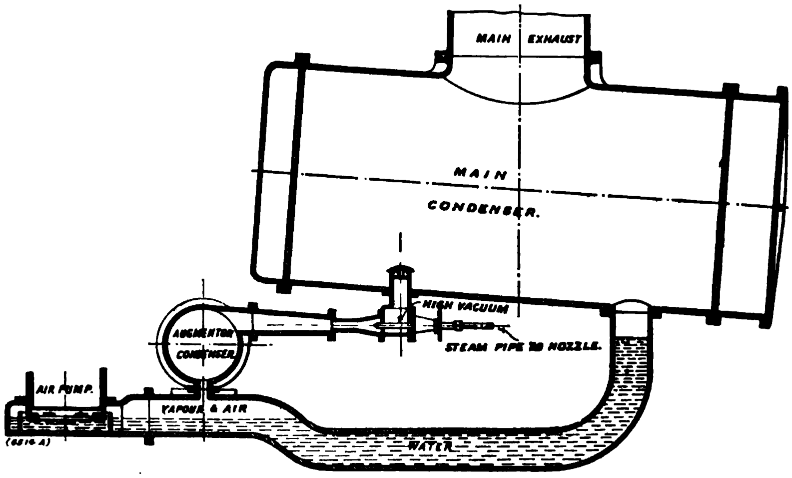


Fig. 95.

indicated. The volumetric capacity of the steam jet is very great, and unless the air leakage is serious, a vacuum a little below the theoretical can be continuously maintained in the main condenser. The steam supply to the nozzle amounts to about $1\frac{1}{2}$ per cent. of the total steam condensed, but this is generally more than offset by the advantage derived from the increased vacuum.

Jet condensers of any type are unsuitable for the production of high vacua, since a large quantity of air is carried in with the condensing water. Hence, to maintain the same vacuum as a surface condenser the air pump has to withdraw an enormously augmented volume of air. In short, no form of condenser is suitable for the production of high vacua in which the condensing water mixes with the condensate.

CHAPTER XXIII.

THE DE LAVAL STEAM TURBINE.

THE De Laval steam turbine was introduced very shortly after that of the Hon. C. A. Parsons, and remains to this day the simplest of any. It embodied several features which have since been incorporated into other designs, such as the use of converging-diverging nozzles, of reduction gearing, and the plan of running the rotor much above its critical speed.

The turbine in its original form consists of a single high-speed impulse wheel, driven by the steam supplied by a series of nozzles arranged round its periphery. It is the only turbine in which fine clearances are not necessary to efficiency, and from the outset has been built on thoroughly mechanical lines, and has probably suffered less from blade stripping than any of its competitors or successors. With the larger sizes, however, cases of buckets breaking loose have not been uncommon, but the buckets are easily renewed. In small sizes the type is relatively economical, but for very large outputs compounded turbines yield better results.

The principal characteristics of this turbine are well shown in the illustrations, Figs. 96 to 99, which represent one of 225 brake horse-power, erected at the Sladen Wood Mills of Messrs. Fothergill and Co., by Messrs. Greenwood and Batley, of Leeds.

The turbine shown is arranged for rope transmission, and was designed to work with steam at a pressure of 200 lb. per sq. in., and superheated to a temperature of 500 deg. Fahr. The rope drive is clearly shown in Figs. 96 and 97. The power is transmitted from the pulleys on the turbine by 28 cotton ropes, $\frac{7}{8}$ in. in diameter, to a second-motion shaft running at 260 revolutions per minute. The wheel speed of 10,000 turns per minute is reduced by means of gearing, which forms an integral part of the machine. To balance the thrust of this gearing, the turbine spindle transmits

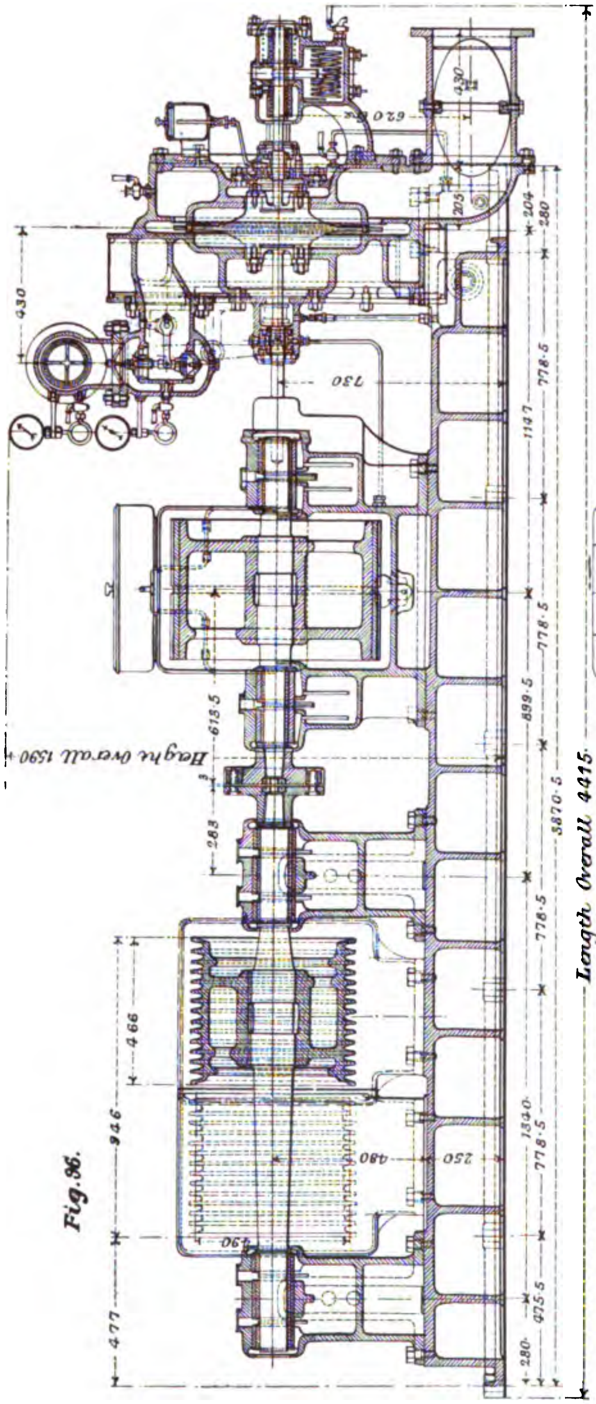

Length Overall 4415
Height Overall 1590

Fig. 96.

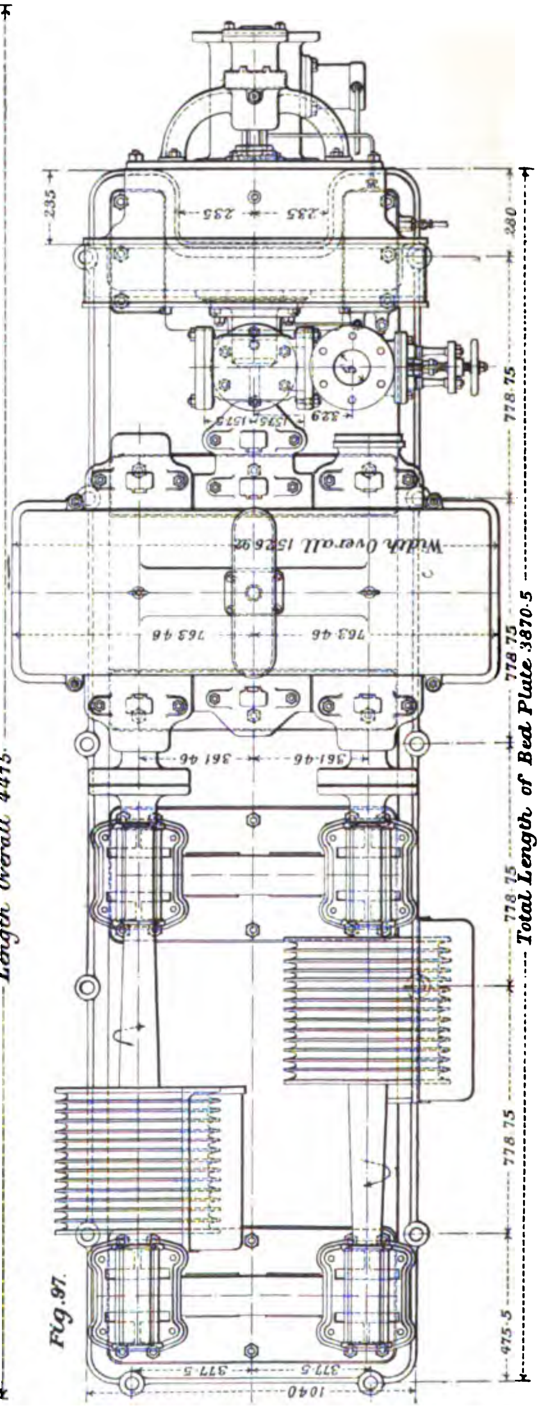

Width Overall 1526.92
Total Length of Bed Plate 3870.5

Fig. 97.

motion to a pair of shafts, one on each side, and in the present case each of these shafts carries, as shown, a pulley driving half the ropes. This is very clearly shown in the plan, Fig. 97, where the turbine casing is seen at the right-hand end of the bed plate, the large reduction-gear casing nearer the centre, and, extending leftwards from it, the two driving shafts, each with its pulley. Metal guards cover almost half the circumference of the rope pulleys, as shown in Fig. 99, below. Fig. 100, page 212, and Fig. 101, page 213, give a sectional plan and elevation respectively of the turbine. The wheel is made as a solid disc, into the circum-

Fig. 98.

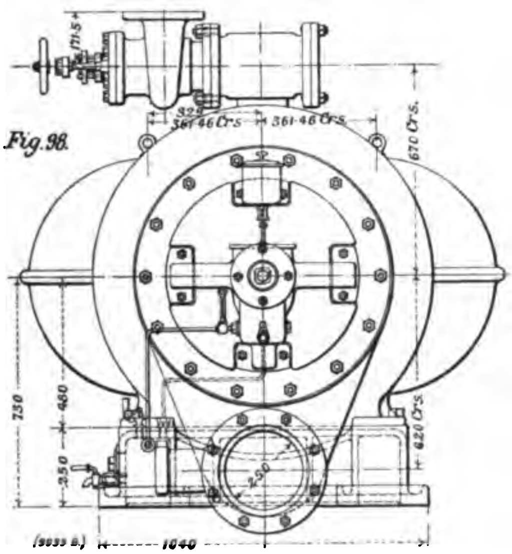
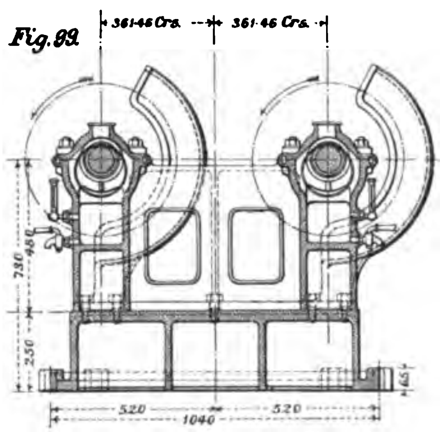


Fig. 99.



Figs. 98 and 99. End Views of De Laval Turbine.

ference of which the buckets are dovetailed, each one being made and fixed separately to the wheel. The steam nozzles, shown in Fig. 102, are arranged at intervals in a ring in close proximity to the turbine wheel, receiving the steam from a steam chest in the turbine casing. Altogether there are eight nozzles, and five of them are provided with shut-off valves, so that they can be closed or opened at any time. This arrangement is of considerable advantage, as, when the turbine is underloaded, some of the nozzles can be closed, and a high efficiency of the machine maintained, even although it is not working at full load. The shaft is made in two pieces, flanged, and secured in recesses on either side of the wheel boss, with steel studs, as may be seen in

Fig. 96. The spherical bearings at each side of the wheel are in two parts, and are arranged to act as stuffing boxes.

The bearings are lined with white metal, and are made as interchangeable bushes. To ensure a satisfactory distribution of the oil a shallow helical groove about $\frac{1}{64}$ in. deep and of $\frac{1}{2}$-in. pitch is

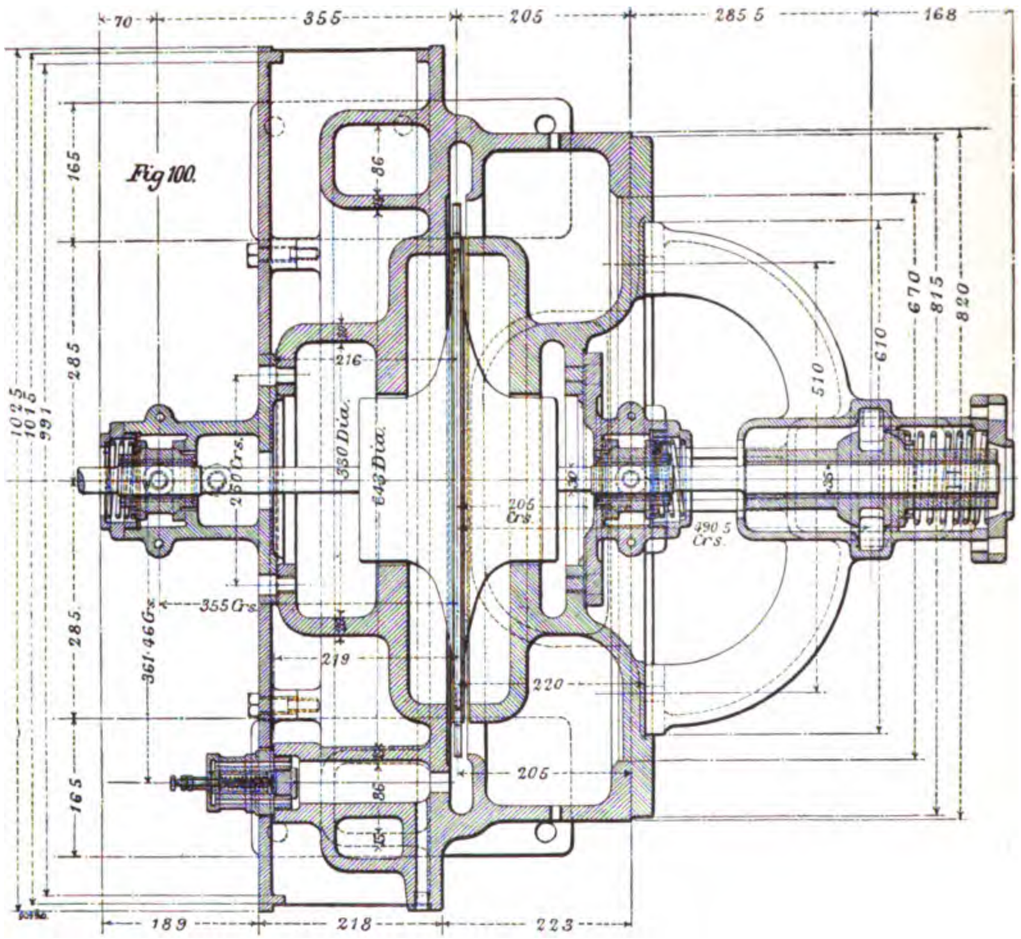


Fig. 100. Sectional Plan of De Laval Turbine.

turned on the interior of the bushes. With the exception of the two bearings supporting the wheel shaft, which are oiled by sight-feed lubricators, all the bearings are lubricated with rings. The oil is filtered, and used over and over again, thus reducing the cost of lubrication to a minimum. The outer bearing, carrying the free end of the shaft, is in one piece, and supported on a cast-iron bracket bolted to the casing cover. In the oil well of this bearing

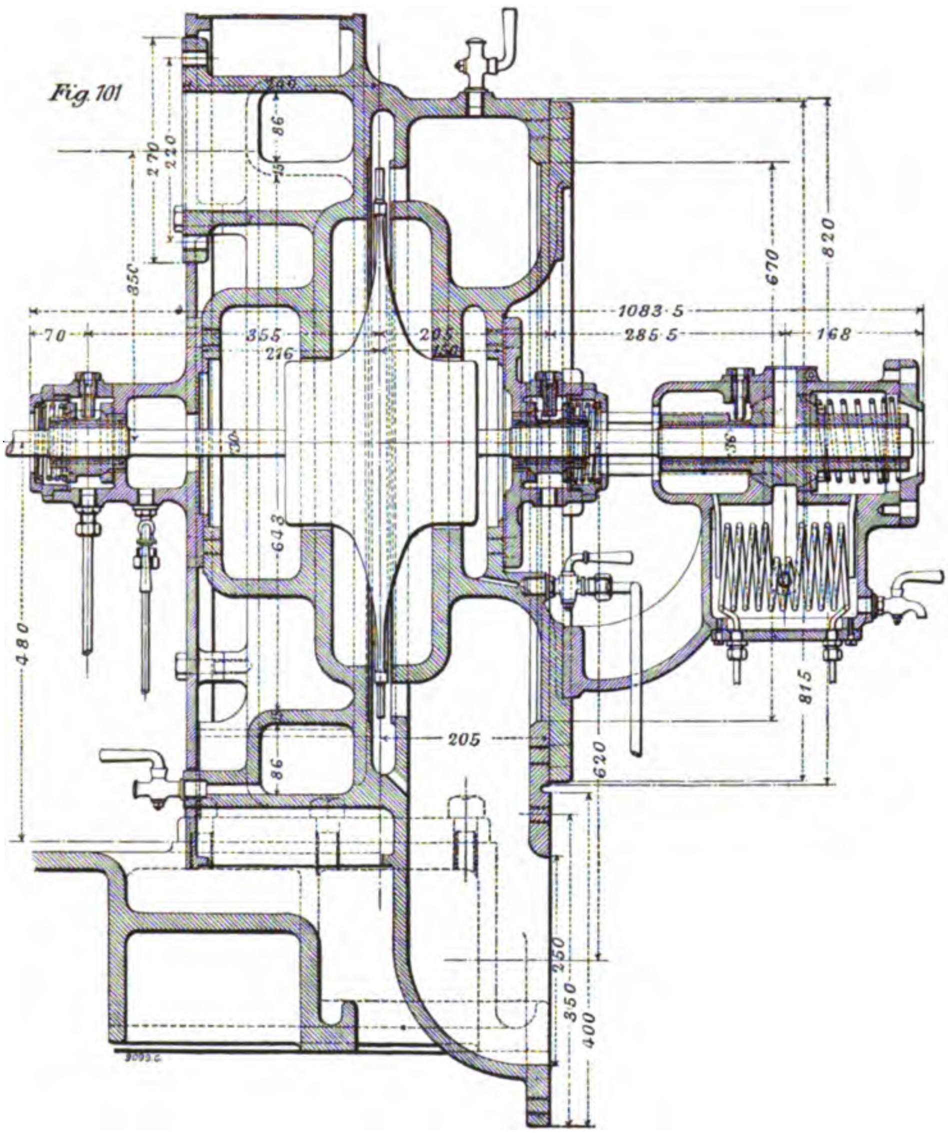


Fig. 101 Longitudinal Section of De Laval Turbine.

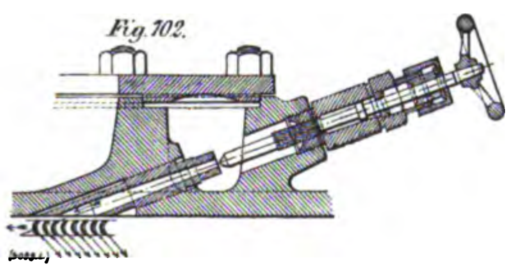
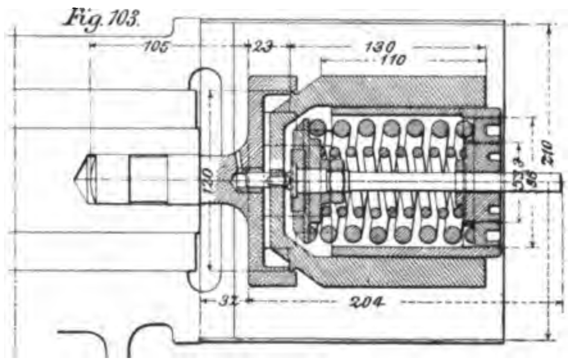
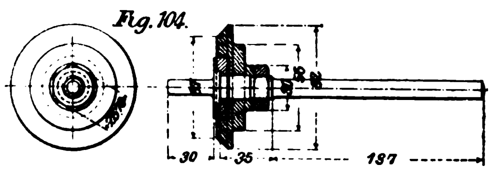


Fig 102. De Laval Nozzle.

is fixed a copper coil, through which water is circulated for cooling purposes. (See Fig. 101.)

The turbine wheel makes 10,000 revolutions per minute, and the speed is reduced to 1000 revolutions by the employment of a helical pinion on the turbine shaft, gearing into two helical wheels on the rope-pulley shafts, as previously mentioned. The pinion is made of hard steel, in one piece, with the shaft and the gear wheels of soft steel of low carbon. The teeth are generated at an angle of 45 deg. with the shaft centre, and the pitch is very small—$\frac{3}{16}$ in.—thus ensuring a smooth contact with very little noise. The



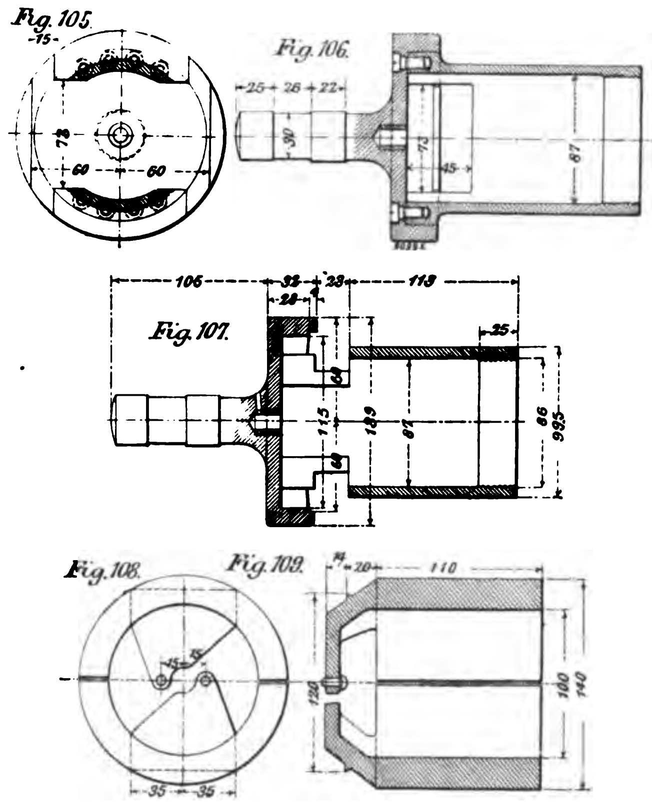


Figs. 103 and 104. De Laval Governor.

gears when examined after some months' use showed no signs of wear, the tool marks still showing on the teeth. The use of such high speeds of rotation involves the solution of some interesting problems in balancing and in withstanding the intense centrifugal forces developed. These are dealt with in the Chapters on Balancing and on the Strength of Discs.

The regulation of the speed of the turbine is effected by a powerful centrifugal governor, the details of which are shown by Figs. 103 to 109, on this and the opposite page. It is fixed on the end of one of the gear-wheel shafts, and controls a double-seated throttle valve, to which it is connected by an arrangement of levers. Fig. 103 gives a section through the governor. The fulcrum piece is a

circular flanged plate, having a stem projecting from the back which is driven into a hole in the end of the reduction shaft. Details of the fulcrum piece are given in Figs. 105 to 107. Attached by set screws to its face is a cylinder, which forms a chamber for the springs. The governor weights are heavy semi-cylindrical pieces, shown detached in Figs. 108 and 109. Instead of turning on pin joints, they swing on knife edges formed across the conical portion at their



Figs. 105 to 109. Details of De Laval Governor.

inner ends. These knife edges are engaged by corresponding grooves on the fulcrum plate, as can be seen in Fig. 103. When the weights fly out, small round-headed pins press against a collar on the central spindle, and move the spindle outwards against the pressure of the springs bearing against the other side of the collar. The spindle fitted with its collars is shown separately in Fig. 104. When the governor is at rest, the walls of the spring chamber prevent inward motion of the weights, so that the latter cannot get away from their fastenings in the fulcrum plate.

The end of the governor spindle when driven outwards by the speed of the turbine comes in contact with a pin at the bottom of the lever shown in Fig. 110. This pin is forced towards the governor spindle by means of a stiff spiral spring enclosed in the boss at the bottom end of the lever. The lever is cottered to a rocking shaft, from which another lever extends upwards, as shown in Figs. 110 and 111. The upper lever is connected to the throttle-valve lever by means of a pin passing through the ends of the two levers, and having knife-edged collars bearing on the bosses. A spiral spring in tension tends to keep the throttle valve open. Sections through the throttle-valve casing are given in Figs. 113 and 114, whence it will be seen that the valve is of the double-beat balance type. The position of the throttle valve, with reference to the governor spindle and rocking shaft, is shown by the dotted outlines of the valve casing in Figs. 110 and 111.

In case of accident to the governor gear a further regulating device is provided in the form of an air valve. This valve, shown in longitudinal section in Fig. 110, and in cross section in Fig. 112, is situated directly opposite the end of the governor spindle, the pin in the lower end of the rocking lever being between. The throttle valve already mentioned will be shut first when the governor spindle protrudes, but should this valve fail to close on account of any accident to itself or the lever motion, the spindle will force the pin in the end of the lever into contact with the emergency air valve. This emergency valve admits air into the wheel casing, which reduces the vacuum. It also lets air into a dash pot, the piston of which is connected by a link to a butterfly valve in the exhaust pipe. The butterfly valve is well shown in Fig. 96, page 210, and on the right-hand side of the turbine casing. It is kept open by the action of a spring, but when air is admitted through the emergency valve, the piston is forced forward and closes the valve against the resistance of the spring. The communication between the wheel casing and the condenser is thus shut off, and the imprisoned steam increases the frictional resistance to the wheel's motion and checks the further expansion of steam in the nozzles.

Figs. 115 and 116 show sections of the steam sieve placed between the stop valve and the turbine casing to prevent any solid matter being carried through the turbine blades by the incoming steam.

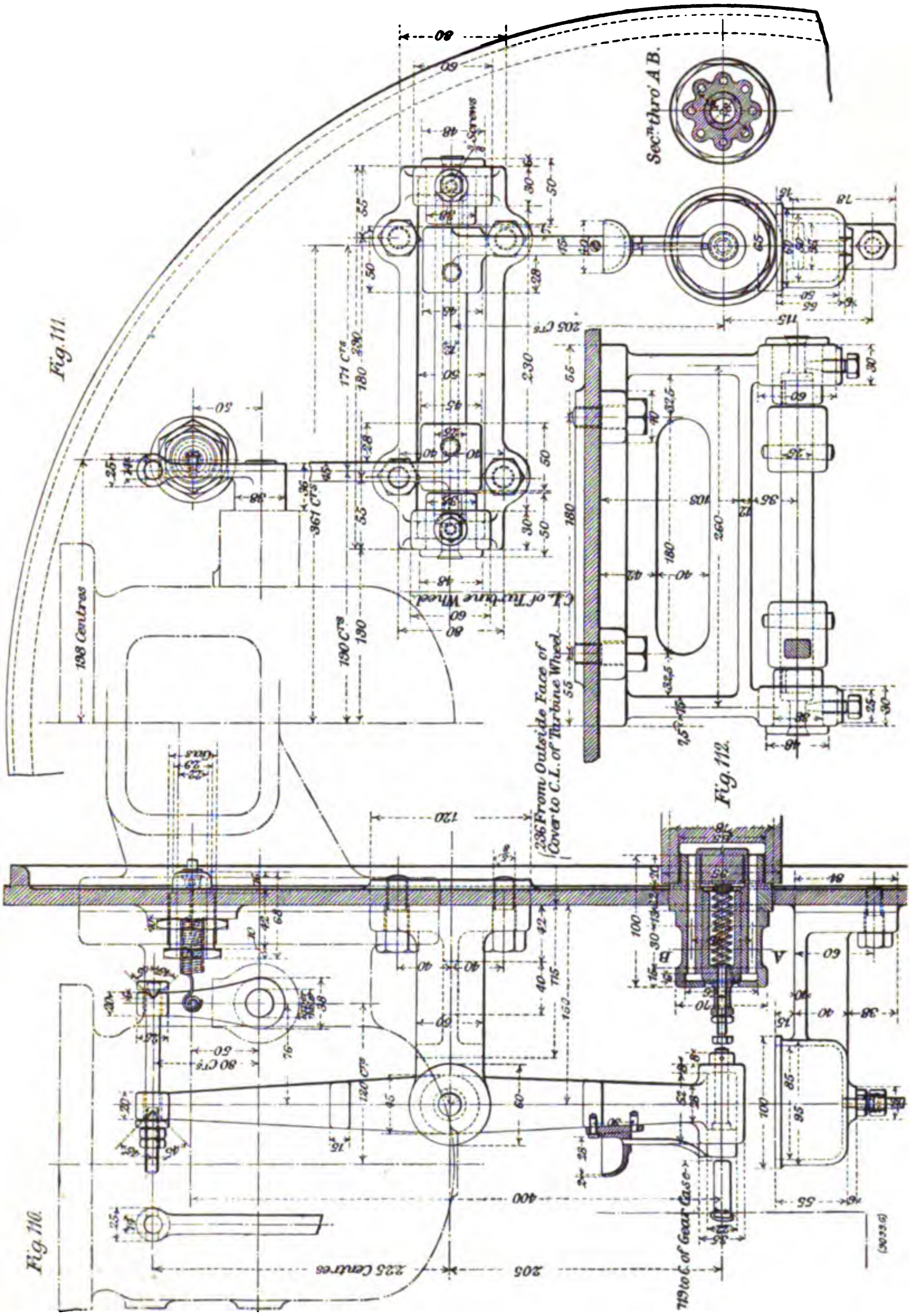


Fig. 110.
Fig. 111.
Fig. 112.
Secⁿ thro' A B.
198 Centres
225 Centres
C.L. of Turbine Wheel
236 From Outside Face of Cover to C.L. of Turbine Wheel
C.L. of Gear Case

It is an iron shell containing a cylindrical sieve of brass gauze, which is held in position by a gun-metal spider. Steam enters the interior of the gauze cylinder, and passes out all round its circumference and downwards through the branch into the turbine.

Fig. 113.

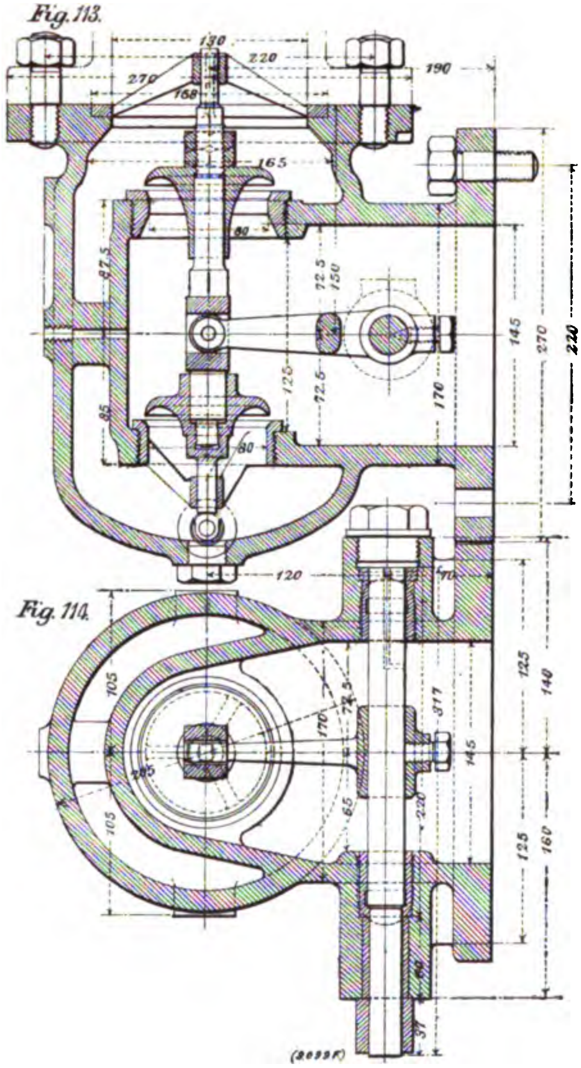


Fig. 114.

Figs. 113 and 114. De Laval Throttle Valve.

The flexible coupling connecting the reduction shafts to the pulley shafts is illustrated in Fig. 117. In the driving flange are eight holes, filled with thick india-rubber bushes. From the other flange project eight studs, which enter the bushes, but are prevented from making actual contact with the india-rubber by thin metal sleeves surrounding them. Small flanges screwed on to the ends

of the studs bear upon the ends of the sleeves, and prevent the coupling coming apart, although allowing a certain amount of longitudinal elasticity.

The following are the dimensions and weights of the principal parts of the turbine:—

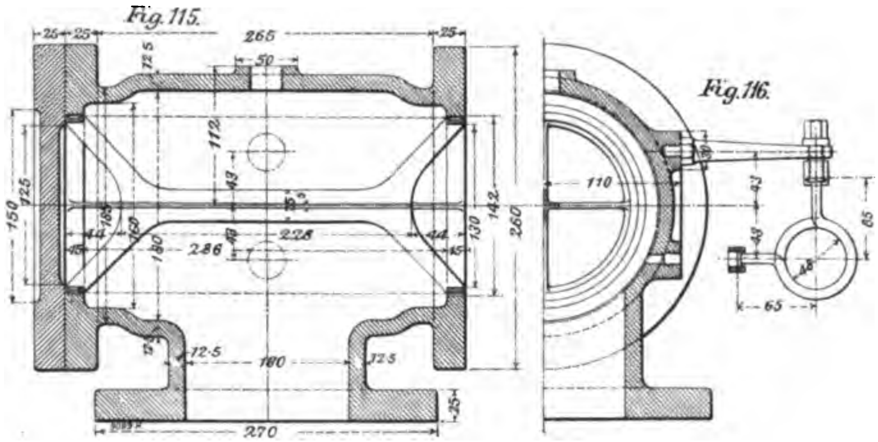
Total weight of turbine ...	7 tons
1. *Turbine Wheel:*	
Diameter in centre of vanes ...	$24\frac{3}{8}$ in.
Outside diameter...	$25\frac{7}{8}$ in.
Weight of turbine wheel ...	$191\frac{1}{2}$ lb.
Weight of one bucket ...	380 grains
2. *Turbine Shaft:*	
Diameter of turbine shaft ...	$1\frac{3}{16}$ in.
Weight of turbine shaft with pinion ...	77 lb.
3. *Gear Wheels:*	
Weight of one gear wheel complete with shaft ...	10 cwt.
Width of face of gear wheel ...	$16\frac{1}{2}$ in.
Pitch diameter ...	26 in. (Gearing, double helical)
Angle of teeth ...	45 deg.
Number of teeth...	296
Normal pitch ...	$\frac{3}{16}$ in.
4. *Pinion:*	
Pitch diameter ...	$2\frac{3}{8}$ in. (Double helical)
Number of teeth...	27

The pitch, breadth, and angle of the teeth are, of course, the same as those of the gear wheels.

The condensing apparatus consists of a surface condenser of 600 sq. ft. of cooling surface, double-acting circulating pump, and a single-acting air pump. A tank below the floor forms the hot well. A feed pump is also provided and connected to the tank for supplying the boiler. The pumps make 120 revolutions per minute, and are worked by a three-throw crank shaft driven by three cotton ropes, $1\frac{1}{4}$ in. in diameter, from the line shaft in the shed. All the bearings are lined with white metal, and lubricated with solidified oil on the Felt system.

The most characteristic feature of the De Laval turbine is its shaft. This is purposely made very slender in proportion to its length, varying in size from $\frac{1}{4}$ in. in the case of a 5 horse-power turbine, $\frac{1}{2}$ in. in a 30 horse-power turbine, $\frac{7}{8}$ in. in a 150, up to $1\frac{1}{8}$ in. in the case of a 300 horse-power machine. In all cases the shaft is

run much above its critical speed. Hence at starting up the wheel runs unsteadily till this critical speed is past. It then "settles down" and runs absolutely steadily, taking up such a position that its centre of gravity coincides with the axis of rotation. It is



Figs. 115 and 116. Steam Strainer.

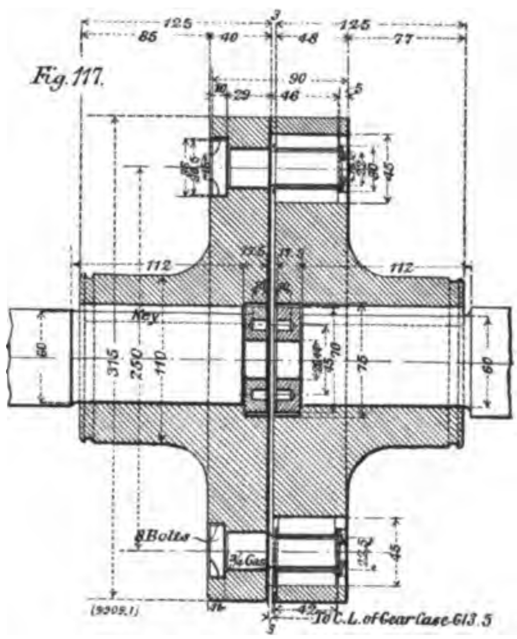


Fig. 117. Flexible Coupling.

possible to run a wheel with as much as two buckets out of balance. The wheels are always made about 50 per cent. weaker immediately under the rim than elsewhere. In the case of a run away, therefore, the rim flies off, carrying the buckets with it, thus eliminating the drive on the wheel. If a portion only of the rim goes the balance

of the wheel is destroyed and its hub extensions foul the holes in the casing, of which they normally rotate quite clear. The following figures give standard proportions of turbines of this type:—

Size of Turbine.	Mean Diameter of Wheel in Inches.	Revolutions per Minute.	Peripheral Speeds in Feet per Second.
5 horse-power	4	30,000	515
15 "	6	24,000	617
30 "	8⅞	20,000	774
50 "	11¾	16,400	846
100 "	19¾	13,000	1115
300 "	30	10,600	1378

The buckets require renewal periodically, as they wear out at their inlet edges, but this is easily and cheaply effected.

The type is in small sizes relatively economical, an efficiency ratio of 24 per cent. having been obtained with a unit of but 10 horse-power, working non-condensing. A 30 horse-power unit, taking saturated steam at 103 lb. absolute, gave a consumption of 39.6 lb. per brake horse-power. Superheating the steam to 500 deg. Cent. diminished the consumption to 25.7 lb. per brake horse-power.

Some results recorded with larger sizes are given below:—

TABLE XVII.—The Steam Consumption of De Laval Steam Turbines.

Horse-power.	Initial Pressure. Pound per Square Inch above Atmosphere.	Steam Temperature.	Vacuum.	Steam taken per Brake Horse-power per hour.
		deg. Fahr.	per cent.	lb.
50	128	—	84	23
100	154	402	83	18
150	107	698	91	14¼
150	113	493	84	17
150	101½	545	94.4	15¼
250	118½	—	93	14
300	194	454	88	13⅞
300	123	378	90	15½
330	138½	563	94	12½

CHAPTER XXIV.

VELOCITY-COMPOUNDED TURBINES.

WITH velocity-compounded turbines the steam issuing from a nozzle, with the velocity corresponding to the total available head, passes without further change of pressure through a series of fixed and moving buckets. No attempt is therefore made to abstract the whole momentum of the steam jet in a single set of such buckets; but the steam leaves the first row of them with a large remanent energy, and is deflected on to a second wheel by a set of fixed buckets, and to this second wheel it delivers up most of the energy carried over from the first wheel. With this system of "velocity"-compounding the reduction of bucket speed is directly proportional to the number of velocity stages. The practical application of this system is mainly attributable to Mr. Curtis.

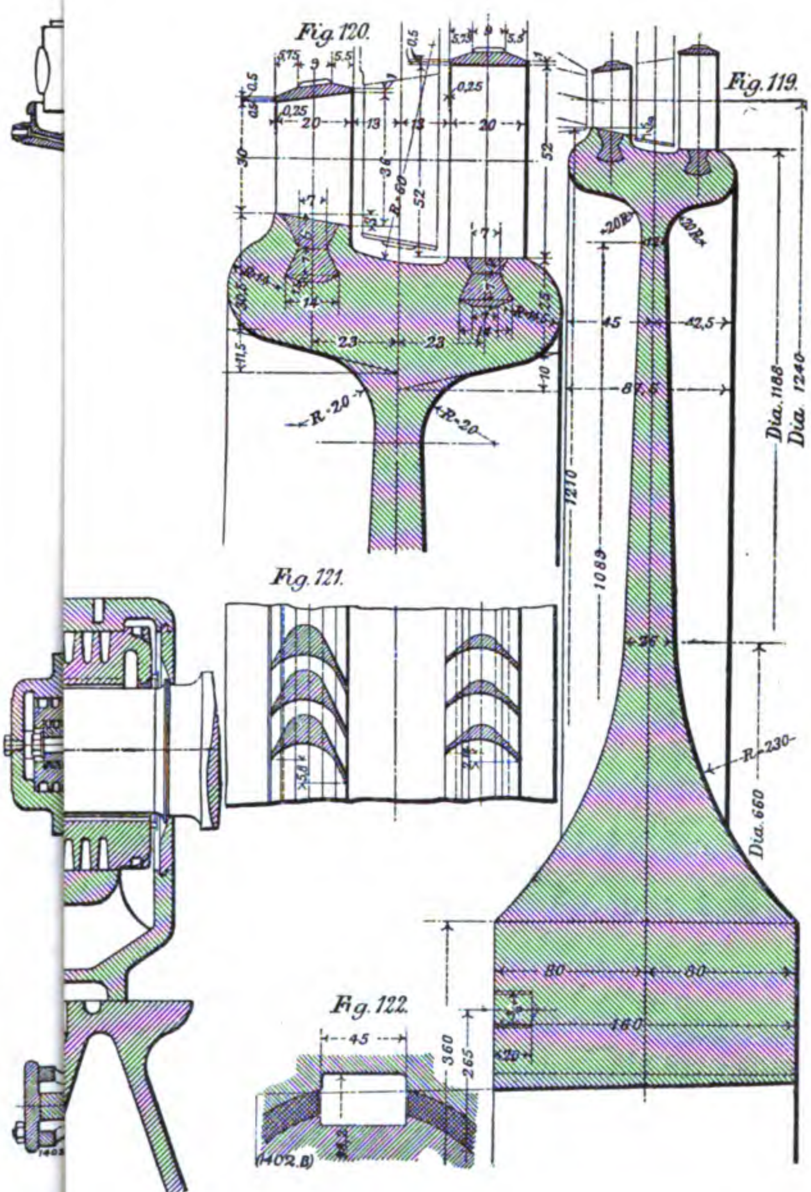
A turbine of this type, constructed by the Allgemeine Electricitäts Gesellschaft, Berlin, is represented in section in Fig. 118. This section, it should be noted in passing, is a bastard one, the steam-inlet valves shown at the bottom being really at the side, as will be seen in Fig. 129, page 226.

As shown, the turbine consists of a chamber divided into two by an intermediate diaphragm, and closed at the front end by a cover, to which are secured the high-pressure nozzle plate and the steam-admission valves. The main chamber is made in halves, which are brought together to form a horizontal joint, thus admitting of the diaphragms being easily fixed in place. The cover is in one piece, and is a steel casting, the rest of the casing being of cast iron.

The construction of the revolving wheels is well shown in Fig. 118, whilst further details to an enlarged scale are represented in Figs. 119 to 122. Each is a solid disc of forged steel, thickened at the centre, where the stresses due to the centrifugal forces are

TURBINE.

BERLIN.



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a maximum, and kept as light as possible towards the rim, in which locality material adds little to the strength, and much to the forces tending to disruption.

The hub is bored conical to a taper of 6 per cent., and between it and the shaft a conical bushing of bronze is provided, which can be tightened up by an end nut, as indicated in Fig. 118. The bush is, moreover, slotted to take a steel key, through which the drive is transmitted. This key, as indicated in Fig. 122, extends through the bush into the shaft below and into the hub of the wheel above. The $\frac{3}{4}$-in. tapped hole, shown in the boss (Fig. 119), is intended to take an eye-bolt should it become necessary to remove the wheel from its seat.

The buckets are of bronze, or of a special make of nickel steel. They are drawn to the forms indicated in Fig. 121. After cutting to length, they are shaped to fit the double dovetailed grooves shown in Fig. 120, and are finally secured in place by caulking in between them spacers of soft brass. The fixed buckets are placed, of course, between the two moving rows, as indicated by the dotted lines, and are mounted on special plates, bolted to the casing.

The nozzles are either of bronze or of a special grade of cast iron. In the former case they have a rectangular section at end, allowing them to be fitted close together. When of cast iron the nozzles are cast in one piece with the nozzle plate. The nozzles extend all round the wheel in the case of the low-pressure section, but over about one-eighth of the circumference only in the case of the high-pressure end. In order to prevent any possibility of a difference in steam pressure being established between opposite sides of the low-pressure wheel, a special port, shown in the upper half of the intermediate diaphragm, is provided, as shown in Fig. 118. This establishes a free communication between the steam on opposite sides of the wheel, and allows any minute disturbance of equilibrium, which may conceivably arise through a sudden change in the load and consequent cutting off of the steam supply, to be equalised, without necessitating a flow of the steam radially across the clearance space. Quite a small pressure difference between the two sides of the wheel might overload the thrust block, but the provision of this equilibrating port prevents any possibility of such a state of affairs.

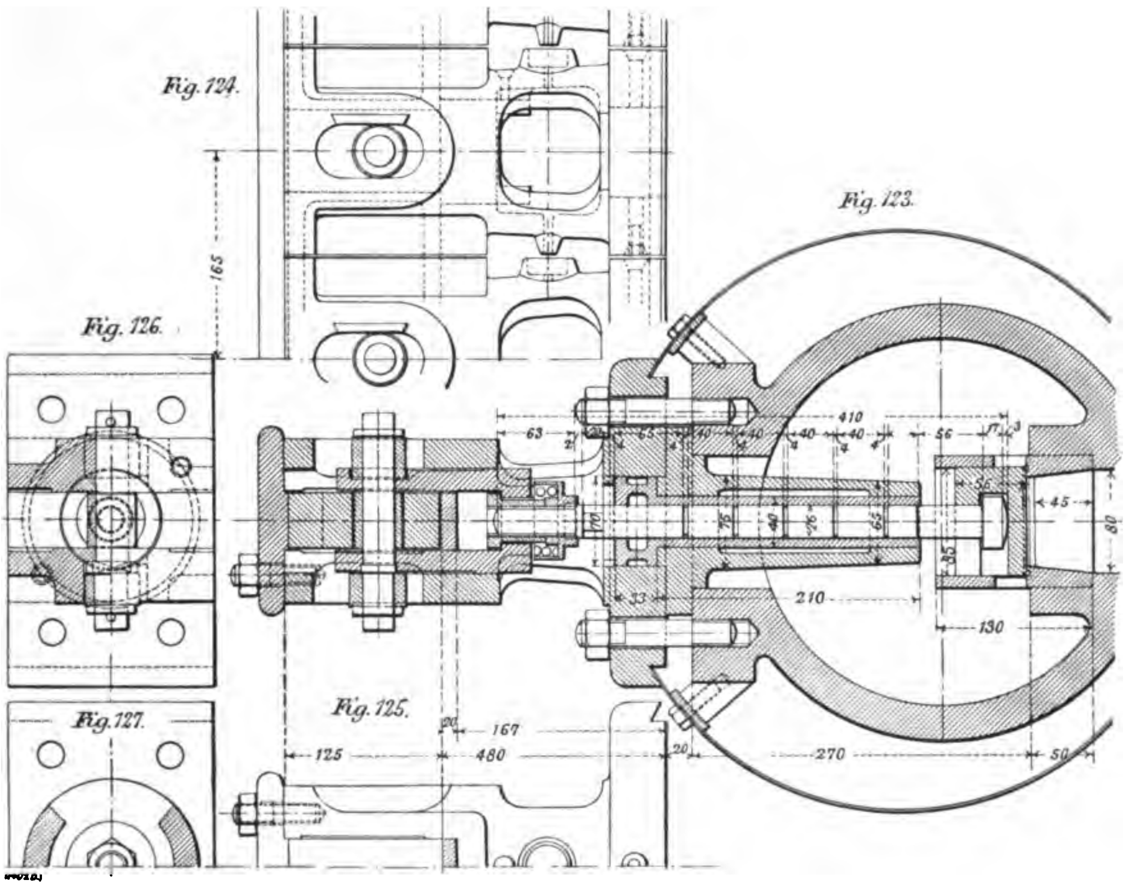
Tho high-pressure nozzles take their supply from a steam chest bolted to the end cover of the turbine. Single nozzles or groups of nozzles are fitted with independent cut-off valves as indicated in Figs. 128 and 129, page 226. The valve spindles are ground to gauge, and work in long sleeves, which are practically steam-tight. A cam plate (see Fig. 128) is moved up and down, as the load varies, by an oil-operated relay controlled by the governor. Pins borne by the outer ends of the valve stems engage with different portions of the cam plate, and as the latter moves up or down, the nozzles are cut in or out in succession as required. The position of the cam plate is regulated by an overtaking motion of the usual type, which will be readily understood on reference to Fig. 129. The valve controlling the supply to the relay chamber is operated by a lever, one end of which is moved up by the governor, and the other end down by the cam-plate stem. Should the governor rise, the lever moving up opens the valve, and a supply of oil under pressure is admitted to the top of the relay cylinder, whilst at the same time the lower side of the piston is opened to exhaust. The cam plate accordingly moves down, and in doing so carries with it the floating lever and closes the valve again, thus cutting off the top of the relay cylinder to pressure, and the underside to exhaust.

Some details of the nozzle-valve gear are shown to a larger scale in Figs. 123 to 126, and sections through the relay cylinder in Figs. 130 to 132. The spring represented in Figs. 130 and 131 ensures that the piston shall be raised when the oil pressure is off, as it is in starting up the turbine.

The governor spindle is driven by spiral gears from the turbine spindle in the usual way. A small thrust block, fitted with forced lubrication, and water-cooled externally, is arranged at the end of the turbine spindle, as indicated in Fig. 118, and maintains the desired clearances between the nozzles and the moving buckets. These two parts, being subject to wear, are made easily accessible for replacement by being mounted at the end of the shaft outside the main bearing.

Figs. 134 to 137 show details of the governor standard, and will be readily intelligible by reference to the key plans given in Fig. 133. The standard is bolted direct to the outside of the end

bearing, both being fitted on the main end cover, as shown in Fig. 118, Plate III. A cover, best seen in Fig. 135, permits of ready inspection of the gears when desired. Forced lubrication is provided for these and for the various journals, the spent oil being collected and passed through a cooler before it is returned again to the bearings. The oil-circulating pump is of the gear-wheel type, and is arranged



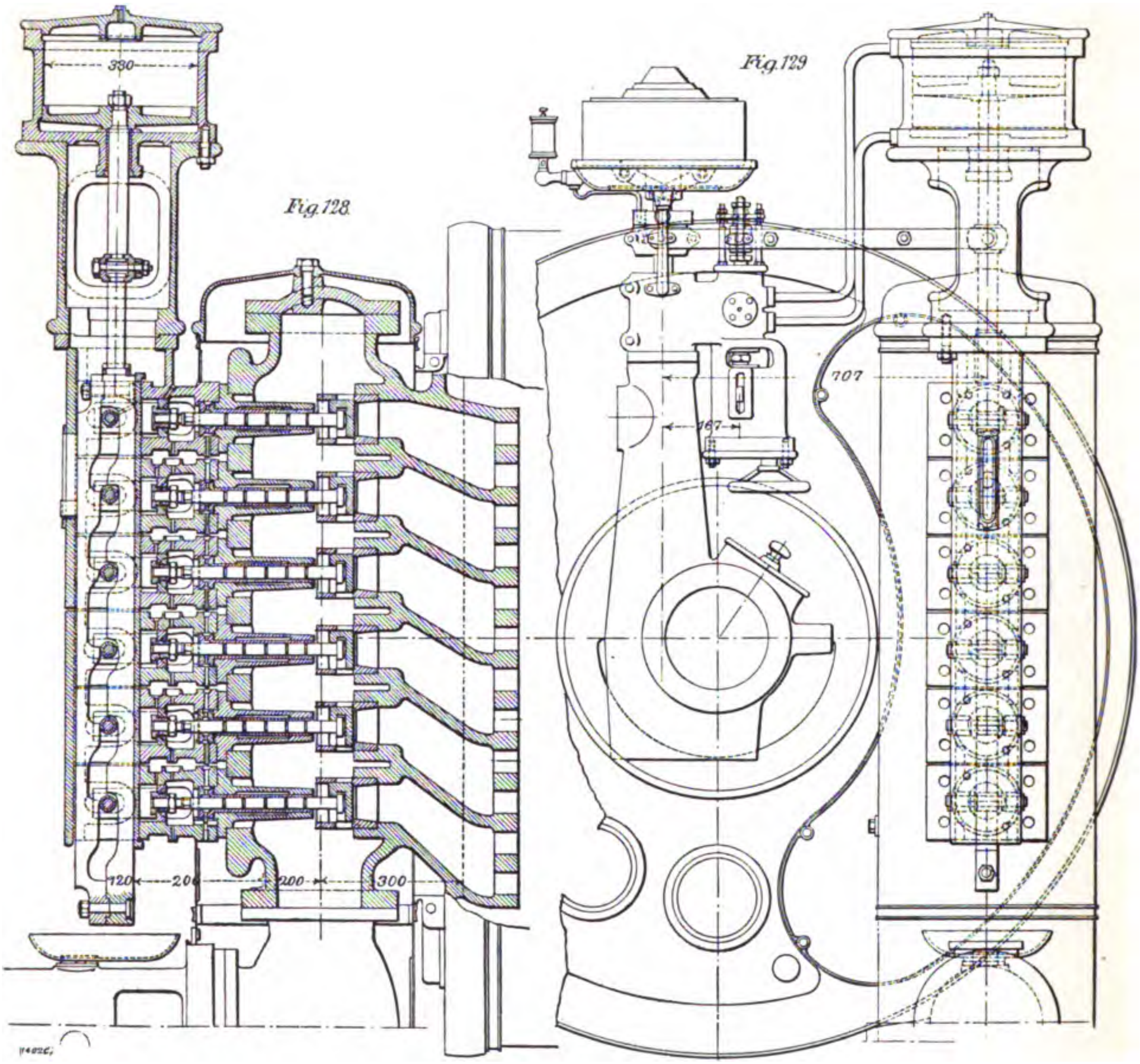
Figs. 123 to 127. Details of Nozzle-Regulating Gear.

at the bottom of the governor standard, being driven from the foot of the governor spindle. Sections through the upper bearing of the spindle are represented in Figs. 136 and 137.

The flange shown to the left in these figures takes the tachometer, which is driven by the bevels, as indicated in Fig. 137. The valve which controls the supply of pressure oil to the relay cylinder is bolted to the top of the standard, as best seen in Fig. 129. The oil exhausted from this cylinder is caught by the annular catcher,

shown in Fig. 137, and falls down to the bottom of the standard, whence it passes again into the oil pump.

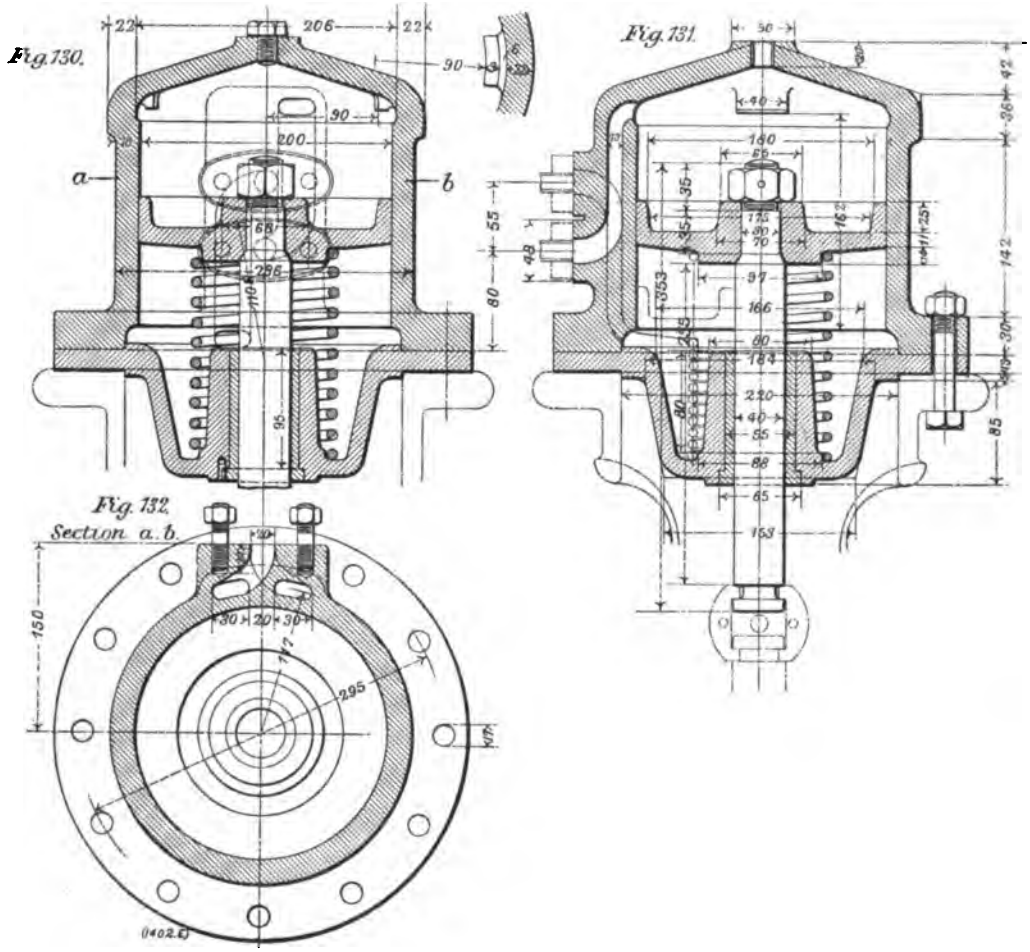
Sections through the plummer blocks at each end of the turbine



Figs. 128 and 129. Nozzle-Regulating Gear.

are represented in Figs. 138 to 141, page 229, and Figs. 142 to 145, page 230. The brasses are, however, missing, but are shown in place in the main section (Fig. 118). The bearing surface is of white metal, oil under pressure being supplied both at the top and the bottom of the bearing, through which it flows axially, and,

escaping over the ends, is caught in the chamber provided at the base of the plummer block, and returned to the oil pump. The efficiency of the lubrication is increased by the circulation of the oil under pressure through a system of ring grooving cut on the outer surface of the brasses, as seen in Fig. 118. In this way the oil is thinned

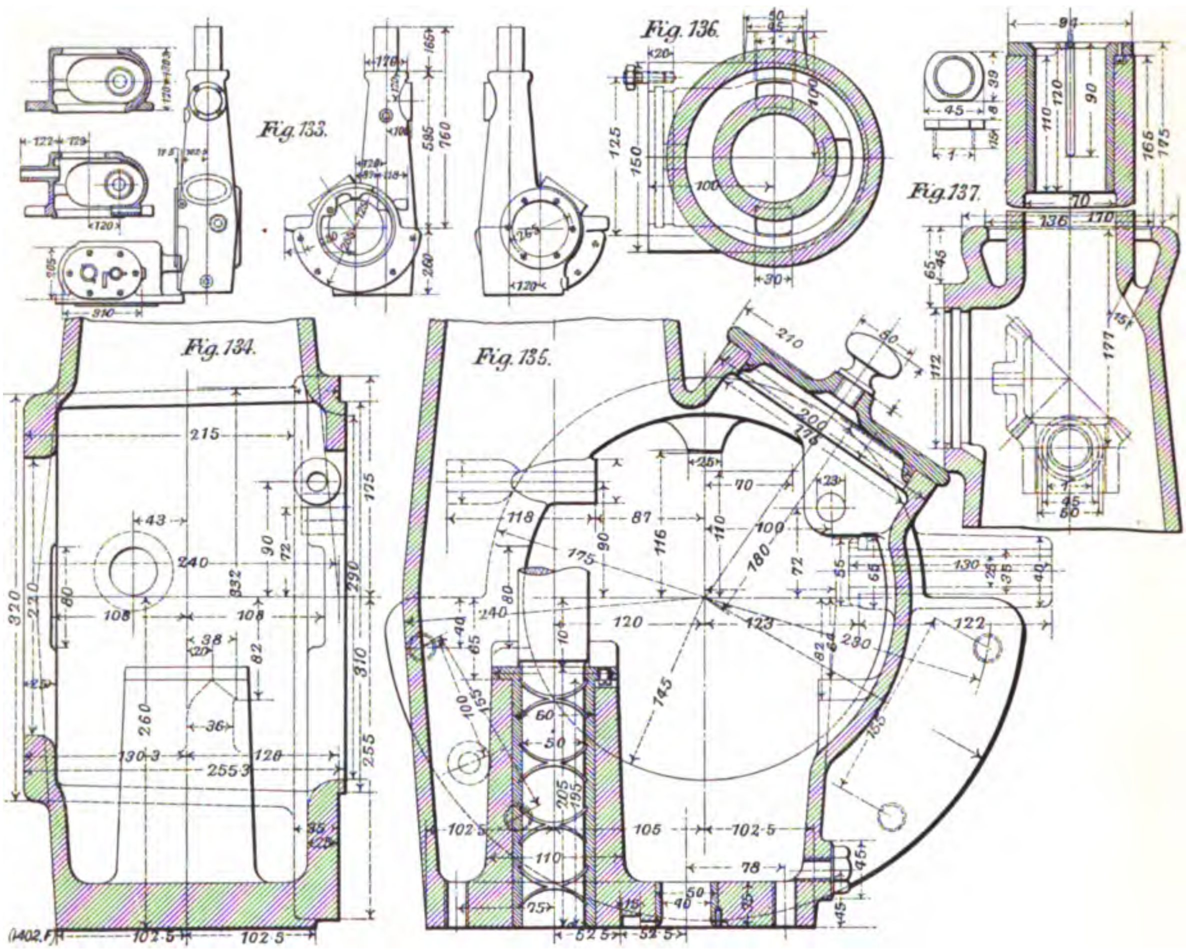


Figs. 130 to 132. Relay Cylinder.

before being passed to the inner surface to be lubricated. The turbine glands are made in halves, and are shown in detail in Figs. 146 to 148, Fig. 149 being a key plan. The packing material in the turbine illustrated is carbon, which requires no lubrication, so that no oil enters the condenser to be carried into the boiler with the feed. But in other cases the A.E.G. also use packings of the labyrinth type.

As another example of a velocity-compounded turbine, we give illustrations of the machinery built for the United States destroyers "Sterrett" and "Perkins," by the Fore River Shipbuilding Company, of Quincy, Mass.

Fig. 150, Plate IV., represents a longitudinal section through

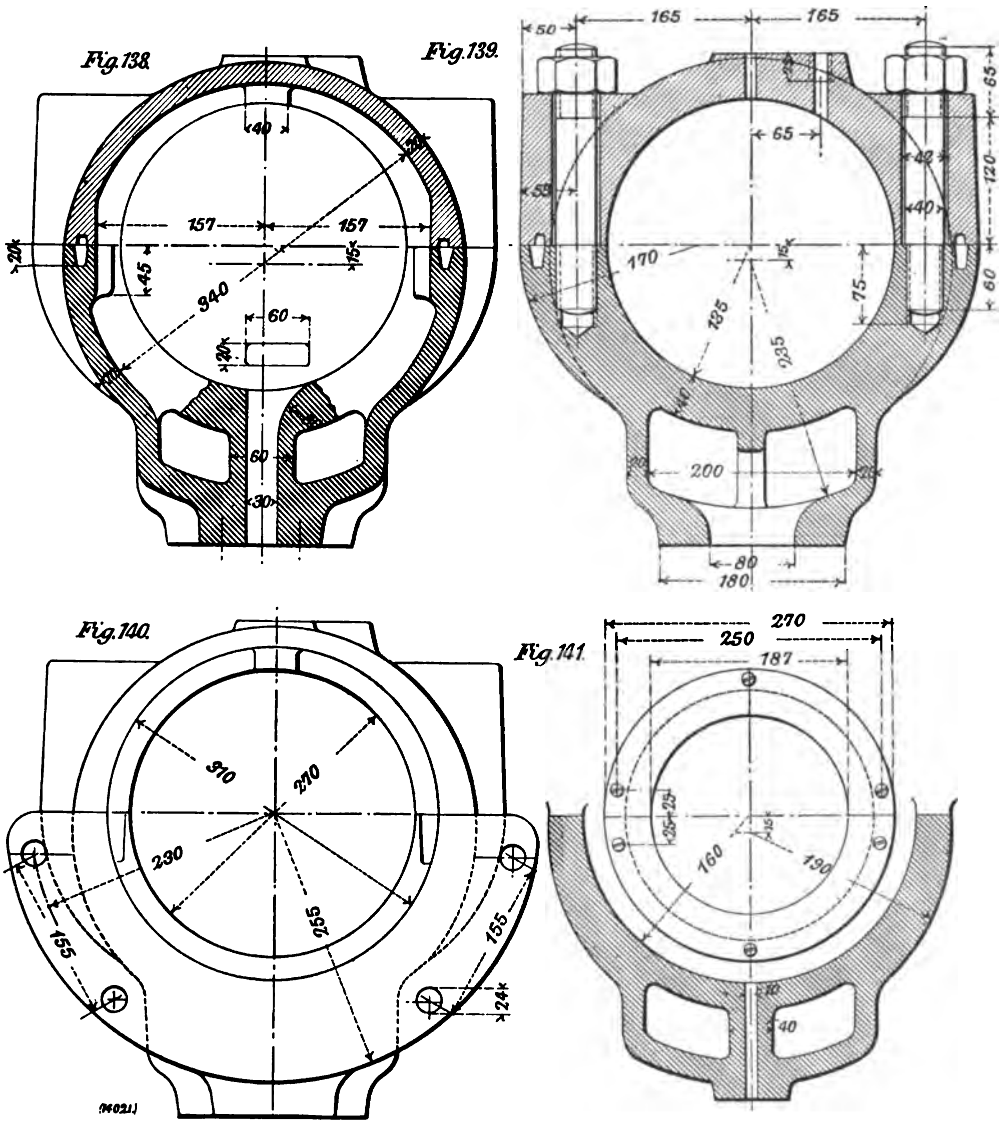


Figs. 133 to 137. Governor Standard.

the turbine, which, it will be seen, consists of fourteen stages, velocity-compounded throughout, even in the case of the blading on the drum section of the rotor. The mean diameter of the turbine-blade path is, in this case, 72 in., and the length of the turbine between centres of bearings is 15 ft.

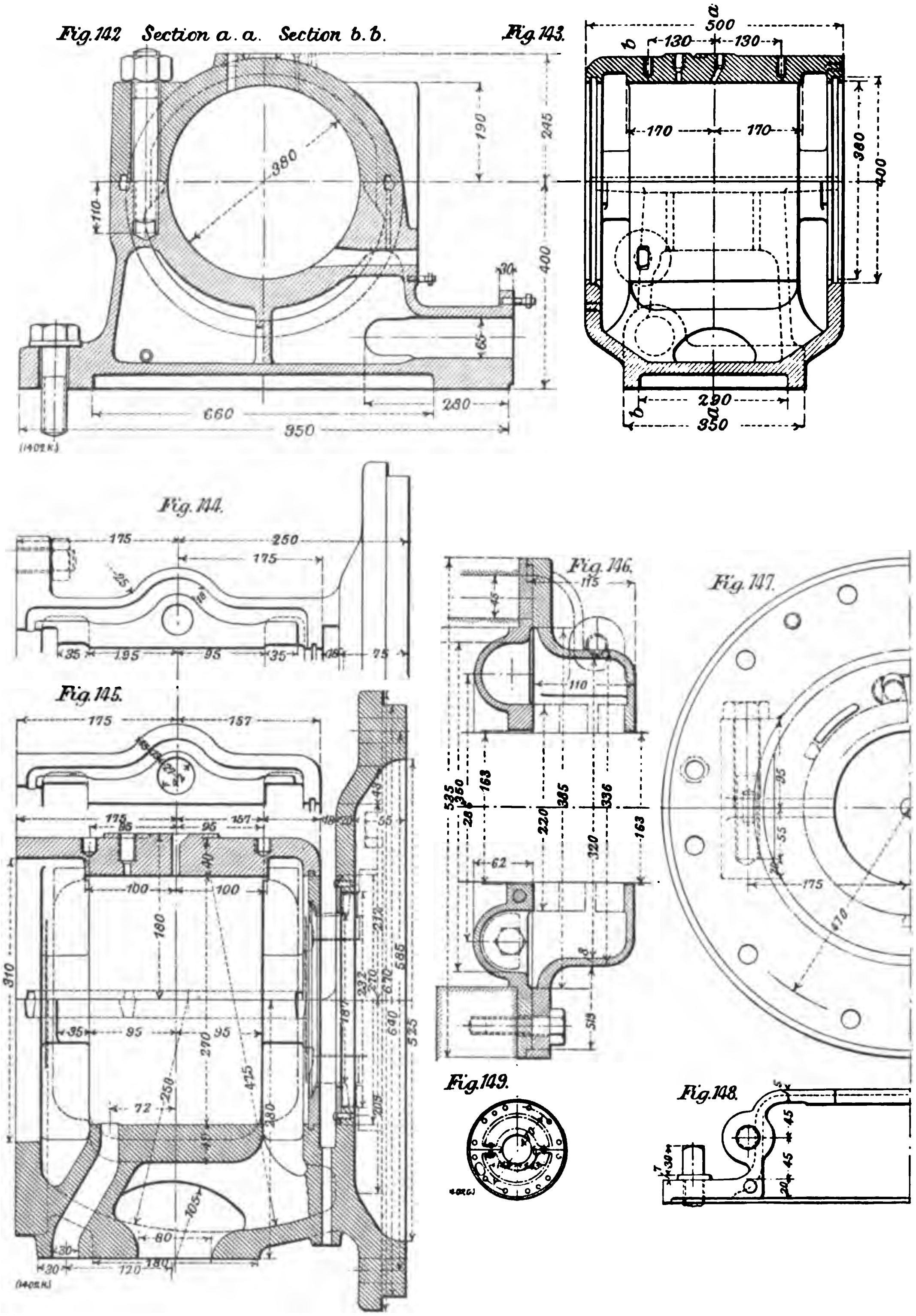
Both the ahead and astern turbines are in the one casing. The latter, which is of hard, close-grained iron, is built up of four

pieces only, consisting of an upper and a lower half, each made in two parts, bolted together by circumferential flanges, as indicated in Fig. 150. The heads are thus integral with the rest of the



Figs. 138 to 141. Main Bearing at High-Pressure End.

casing, and they are provided with lugs, to which are bolted the brackets carrying the main bearings. All the flanged joints are scraped to a fit. The turbine thus forms a unit free from risk of distortion in erection, and easily manufactured in correct alignment throughout. The whole of the interior of the casing is



Figs. 142 to 147. Main Bearing at Exhaust End of Shaft.

NS" AND "STERRETT."

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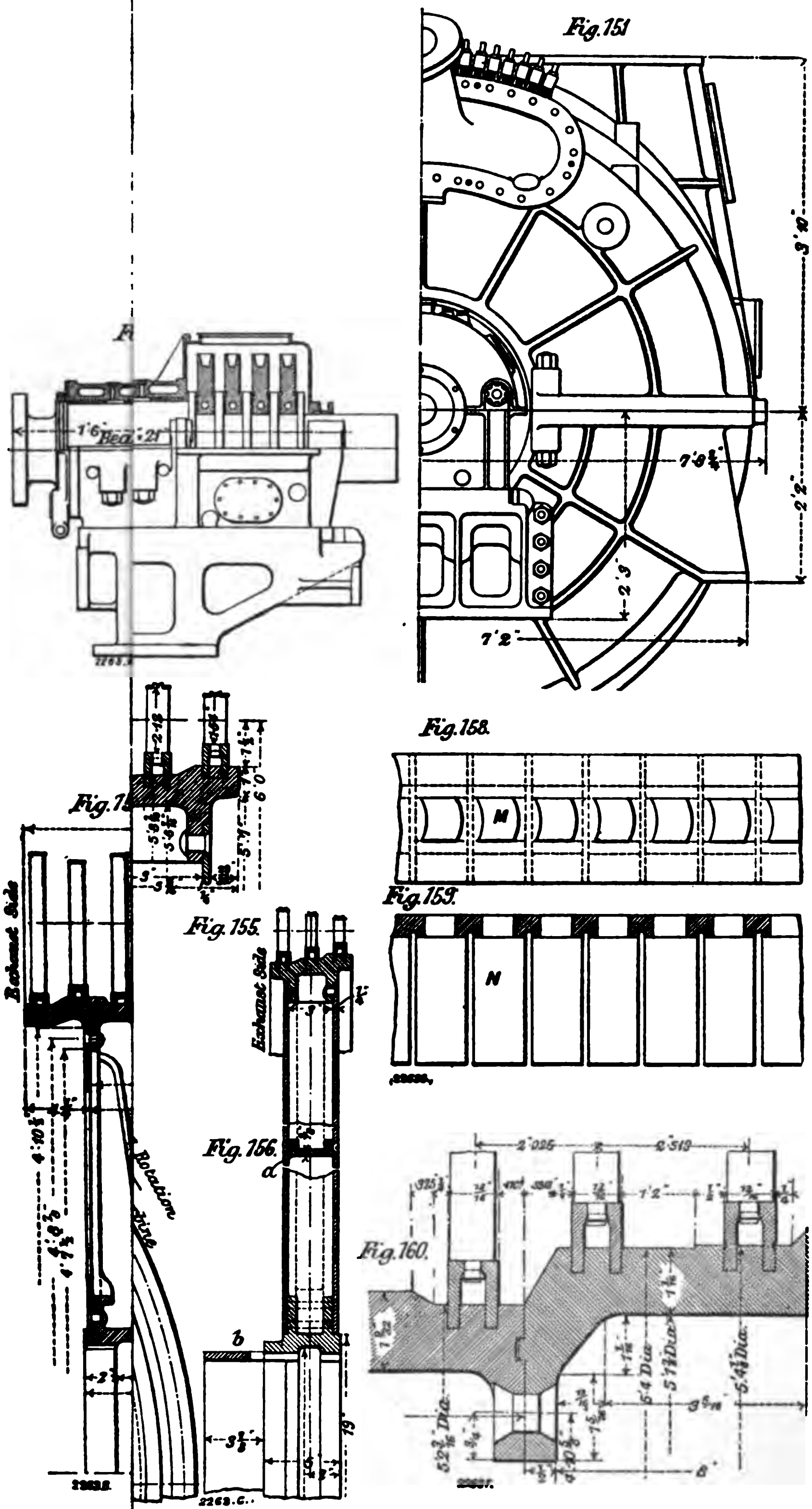

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machined, thus ensuring the easy and accurate fixation of the various nozzle plates and stationary bucket segments. Feet, for bolting the casing to the foundation plates in the ship, are cast on the lower half of the casing, and the exhaust branch is at the top, as shown. The main shaft is a high-grade steel forging, bored from end to end, and is carried on two bearings. The forward end has thrust collars turned on it, and the thrust block is integral with the forward bearing, thus ensuring its correct alignment.

The ahead rotor comprises six wheels for the stages having partial admission, whilst the remaining eight stages are arranged on a drum as shown. The main steam inlet is shown to the right in Fig. 150, and at the top in Fig. 151, and leads into a nozzle box shown separately in Figs. 177, 178, and 179, page 234. This box has nineteen nozzles, of rectangular section, each of which, as shown in Fig. 179, is fitted with an independent shut-off valve. By closing several of the nozzles, the output of the turbine can be reduced without the loss which is involved when the power is controlled solely by throttling the steam. These nozzles, as best seen in Fig. 180, Plate VI., are all formed in a single bronze casting, which is bolted by machine screws (secured from working loose by having the metal caulked into the slots) to a forged and machined steel plate, which fits on to machined seats in the nozzle box, and forms the surface on which the shut-off valves slide, being scraped to a true surface on this space. These nozzles are cut out of the solid after casting; they are slightly divergent; the total throat area of the nineteen provided is 8.892 sq. in., and the area at discharge is 1.145 times as much. As will be seen, no portion of the nozzle is parallel. The centre line of the nozzle makes an angle of 20 deg. with the plane of the wheel. As the shut-off valves may have to stand open for very long periods, provision is made for balancing them when open. To this end a groove, $\frac{1}{8}$ in. wide by $\frac{1}{16}$ in. deep, is cut in the valve plate, as shown at *a*, Fig. 177, and the edge of the valve, when open, overrides this, permitting steam to enter below it, and ease it off its seat. In these nozzles the steam under full-power conditions is expanded from 246 lb. absolute down to 86.80 lb. The theoretical velocity of issue is thus about 2080 ft. per second. These first-stage nozzles deliver the steam on to a velocity-compounded wheel, having four rows of moving buckets,

the mean speed of which in the full-power trial was about 187 ft. per second.

Attention may be called to the method by which the brackets for the nozzle spindles are held in place. As indicated in Fig. 178, page 234, each is located truly by two steady pins, and is then held in position by nuts and washers, each of which overlaps two adjoining brackets, there being a half-hole in the flange of each.

Details of the blading of the first wheel are shown in Figs. 165, 166, and 171, Plate V. The bucket angle of the first row at entrance is, it will be seen, 28 deg., whilst at discharge it is 22 deg. From this it follows that the space between the buckets is narrower at discharge than at entrance, but this is in accord with the fact that the stream of fluid as it passes through the bucket tends to spread laterally, and consequently becomes thinner. It will further be observed that each successive bucket is longer than its predecessor. This is necessary, because in each bucket the fluid, besides thinning, as already mentioned, also loses velocity, and thus a greater steam-way is required at each successive set of buckets. As a consequence, the length of the last bucket is $2\frac{1}{2}$ times the width of the stream of steam as it first issues from the nozzle.

The blades, which are of extruded bronze, are all carried on foundation strips, which are illustrated separately in Figs. 158 and 159, Plate IV., but the method of mounting will be best understood on reference to the reproductions from photographs of the blading, Figs. 175 and 176. The foundation rings are strips of steel, milled to a U-shape, and grooved along both sides, so that they can be secured in place on the wheel rims by caulking the metal of the latter into these grooves. As received from the milling machine, the foundation rings are straight. They are punched, as shown in Fig. 176, page 233, to receive the roots of the blades, which are secured by riveting. To enable the rings to be bent to the curve of the wheel, they are slotted, as shown in Fig. 159, just referred to, and also in Figs. 175 and 176, page 233. The outer ends of the buckets are provided with shrouding, riveted on, as indicated in Fig. 175. The ingenious method of assembling and mounting the blades lends itself to very rapid manufacture, since the construction of the blades and foundation strips can be proceeded with at the same time as the rotor and casing

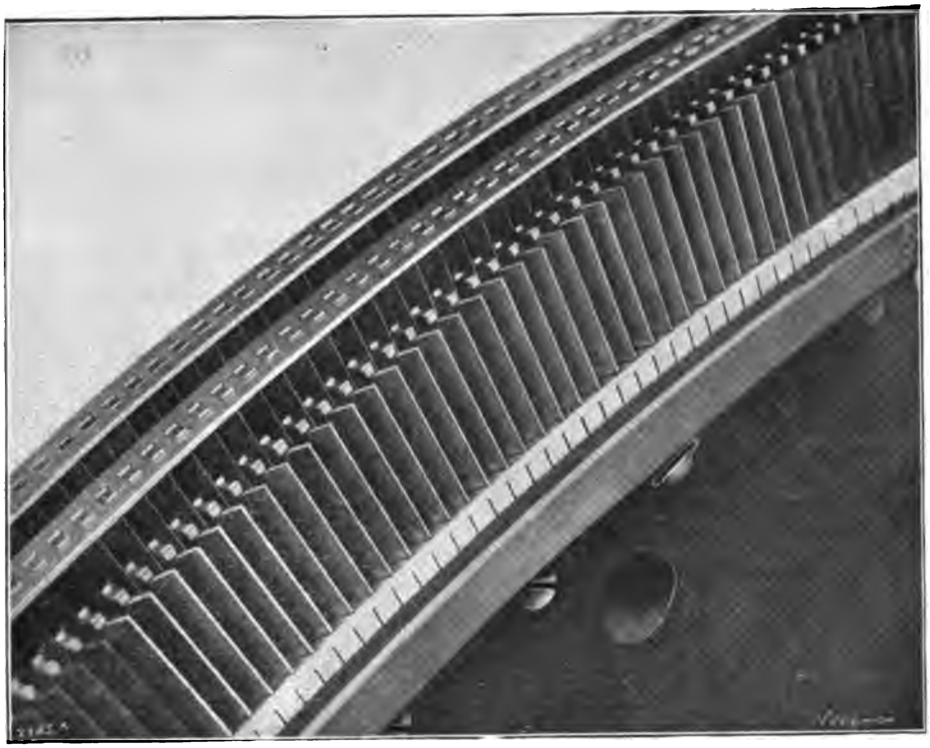


Fig. 175. Blading of Rotor.

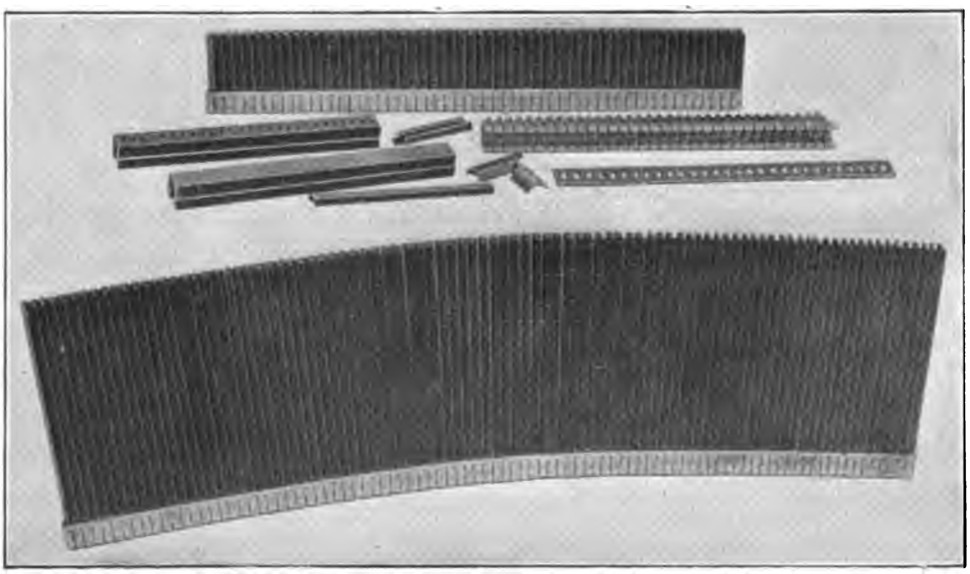
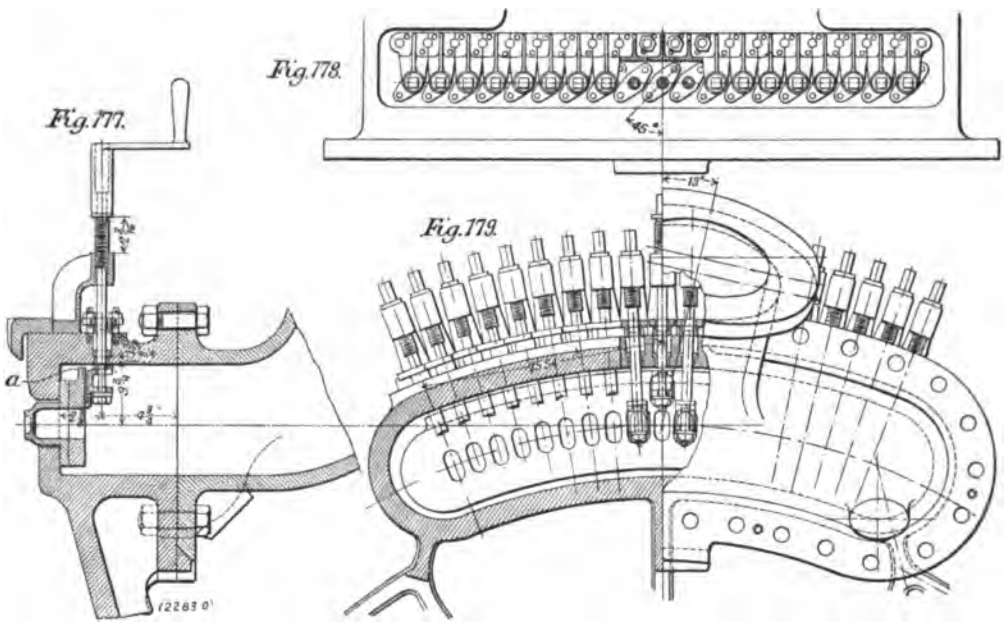


Fig. 176. Blades and Foundation Strips.

are being machined. The method of finally fixing in place is quick and simple, and experience has shown it to be reliable, the shrouded segments being very strong mechanically and very firmly held in place by the caulking

The intermediate or stationary buckets, which are interposed between the different rows of moving buckets, so as to catch the steam discharged from the one row and direct it in a favourable direction on to the next, are also mounted on similar foundation rings secured to steel holders, and are bolted into place. Doors



Figs. 177 to 179. Nozzle Box of Curtis Turbine.

for inspection purposes are provided to the rectangular openings immediately above these blade segments, and through them access can be obtained to the individual wheels, without the necessity of lifting the whole upper half of the casing. A diaphragm (see Figs. 173 and 174, Plate V.), built up of dished boiler-plating, in most cases $\frac{1}{4}$ in., but for the first stage $\frac{5}{16}$ in. thick, riveted to steel castings at the hub and the periphery, divides off each wheel from its successor. At its circumference each of these diaphragms has a rib, which engages with a slot turned in the casing, and makes, when the pressure is on, a practically steam-tight joint. At the centre, where the shaft passes through the hub, the diaphragm is

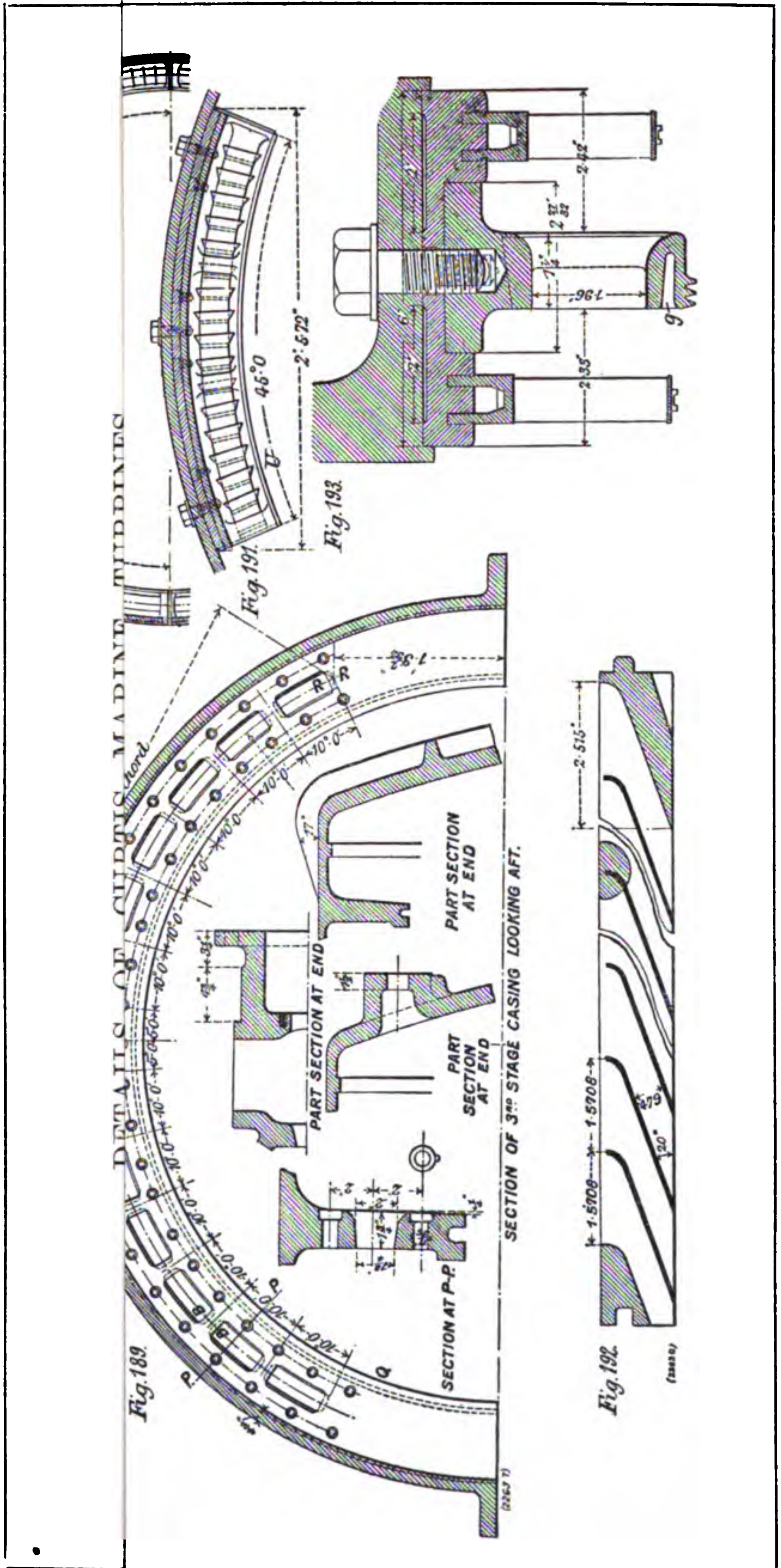


Fig. 189.
Fig. 191.
Fig. 192.
Fig. 193.
SECTION OF 3RD STAGE CASING LOOKING AFT.
PART SECTION AT END
PART SECTION AT END
PART SECTION AT END
SECTION AT P-P

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provided with a bushing of anti-friction metal, as shown to a larger scale in Fig. 172. This bushing, it will be seen, is turned into a series of serrations, so that if a touch occurs between it and the shaft liner there is no sensible production of heat. The clearance, when hot, between this bush and the shaft liner, is 30 mils in the case of the first of these diaphragms, 40 mils for the second, 50 mils for the third, and 60 mils for the remainder. The diameter at the first diaphragm is $16\frac{5}{8}$ in. and increases in successive steps of $\frac{1}{8}$ in. each up to $17\frac{1}{4}$ in. at the last diaphragm. The bushings are held at four points only, and it will be seen that the part nearest the shaft is considerably overhung, thus giving that portion a little elasticity in case of the shaft whipping in consequence of a sudden change of load.

The nozzle openings for each stage after the first are formed by sections of nickel-steel plates fused into cast-iron ribs, as best seen in Fig. 192, Plate VI. These plates are bolted to ribs cast on the main turbine casing, as shown in Fig. 150, Plate IV. The bolts used in all these cases, it may be observed, are fillister-headed machine screws. These have proved amply sufficient, as the parts connected are subjected to a steady load only, and do not require to be held together with any extraordinary degree of pressure. The screws are prevented from unscrewing in service by simply caulking some of the surrounding metal into the slot.

Sections through the casing showing the ribs to which the nozzle plates are screwed are represented in Figs. 188 and 189, Plate VI. As will be seen from the section taken at A A, the opening in the casting is bell-mouthed, so as to give the steam an easy access to the nozzles, and, as the pressure drop in all stages but the first is small, these nozzles are not divergent.

For stages 2, 3, 4, 5, and 6 the wheels carry three rows of moving buckets; their construction is shown in detail in Figs. 155, 156, and 157. They are built up of $\frac{1}{4}$-in boiler plating, riveted to forged-steel rims and hubs. The latter are slotted to take a key $1\frac{1}{4}$ in. wide, and are turned to a diameter $\frac{1}{1000}$ in. less than their seats on the shaft, on to which they are forced by hydraulic pressure. Each hub has further drilled in it four $\frac{3}{4}$-in. holes to take bolts to pull off the wheel, if in any case it should be necessary. Each wheel is held in its axial position by distance

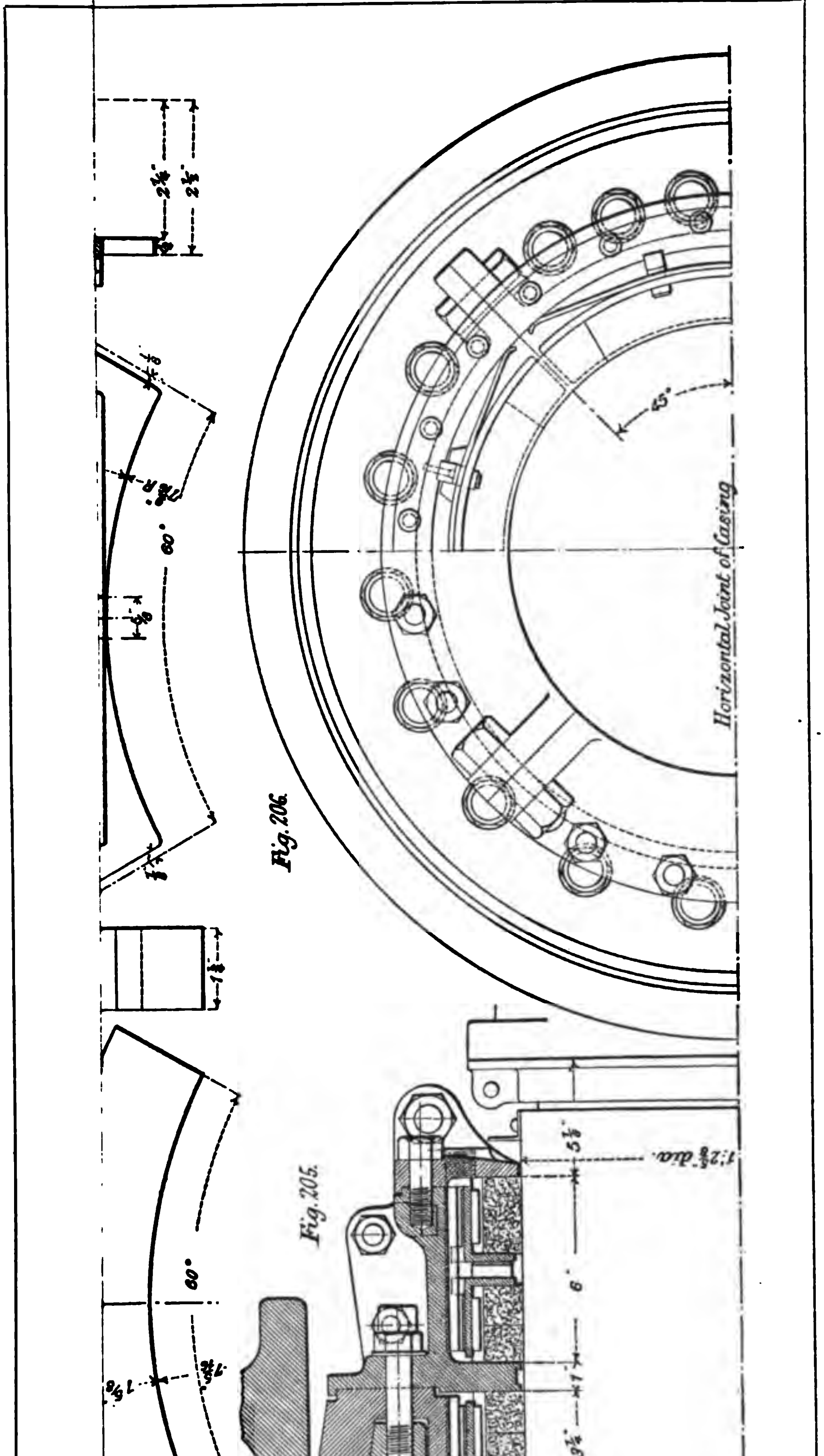
pieces, as best seen in Fig. 150, and the whole rotor is prevented from working forward by a lock-ring, which enters into a groove cut in the shaft, and is secured to the hub of the first wheel.

These distance pieces are checked from turning by the key, which extends beyond the wheel hubs, and fits into a slot cut in the sleeve, as shown at *b*, Fig. 156, Plate IV. The distance pieces are thick enough to mask the keys, so that the outer surface of the shaft is thus rendered quite smooth. The sides of the wheel are stiffened by eight arms or ribs riveted to the hub and slotted into a channel section with a view to reducing the weight, as indicated in the "bastard" cross section at *a*, in Fig. 156. A view of the rim to a larger scale is represented in Fig. 157, whilst details of the blading will be found in Figs. 161 to 164, and Figs. 167 to 190. The fixed intermediate rows of blades are secured as illustrated in Figs. 187 and 193.

All the "wheel" stages work with partial admission, though in the case of the lower pressure stages there are nozzles in the lower half of the casing as well as in the upper. In the case of the drum section of the turbine, however, the admission is complete. A view through the casing in this region is represented in Fig. 190, Plate VI., whilst Fig. 191 shows to an enlarged scale the method of securing in place each nozzle segment, of which eight complete the circle. These nozzle blocks here are bronze castings, and, as shown in Fig. 193, are in the first place secured by machine screws to the forgings, on which the fixed blades are mounted, and each unit thus formed is then fixed to the casing by set bolts.

The holder shown in Fig. 193 is that at the commencement of the drum section, and, as will be seen, it carries the fixed blades for both the seventh and the eighth stages. On reference to Fig. 150 it will be seen that for the succeeding stages each of these holders carries one set of fixed blades only.

Returning to the nozzle segments, these nozzles are formed by casting in division plates, as shown in Fig. 192. Where the edge of the nozzle segment is in contiguity with the rotating drum, it is serrated, as best seen in Fig. 193. The deep slot formed at *g* is for the purpose of giving a little flexibility to the baffle in case of an accidental touch. On this drum section of the turbine there are, it will be seen, two rows of moving blades per stage.



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The construction of the drum is well shown in Figs. 152 and 153, Plate IV. It consists of an eight-armed steel casting plated at both ends. The rim which carries the blading is in three pieces, connected together by rivets passing through internal flanges, as best seen at Fig. 160. This drum is secured to the cast-steel spider by the end plates, which are of steel $\frac{1}{4}$ in. thick, riveted to the rim, the arms, and the hub, and at the after end this plating has large man-holes through it, as indicated by the dotted lines in Fig. 153. The forward end of the drum being closed in, there is on this drum, when the turbine is at work, an end thrust which balances that of the propeller, any residual thrust being taken by the thrust block shown to the right in Fig. 150. Details of the drum-blading are illustrated in Figs. 161 and 162, 167 and 168, and 169 and 170. Figs. 169 to 171 represent the blading for the thirteenth and fourteenth stages. These blades are, it will be seen, some 9 in. long, and are consequently made 1 in. wide instead of $\frac{3}{4}$ in., as in the case of the shorter blades.

Details of the main bearings are represented in Figs. 194 to 196, Plate VII. These bearings have water-jacketed brasses lined with white metal. Lubrication is effected by oil pumped in through the two openings shown in the top half. This oil flows both ways through the bearing. That flowing aft is caught by an oil thrower, and drained away through the opening shown at the left-hand bottom corner of Fig. 194, whilst the oil flowing forward passes into a reservoir into which dip the thrust collars on the shaft, as indicated by the dotted lines.

Cooling pipes (see Figs. 197, 198, and 199, Plate VII.) are provided at the bottom of this reservoir. The surplus oil is drained off to an oil cooler, and, after straining, is returned to the bearings. Details of the fixed thrust collars are represented in Figs. 201 to 204. These are faced with white metal, thoroughly hammered into place. Oil for lubrication purposes is forced in through the openings at k and l, whilst cooling water is circulated through the interior of the collars by the pipes shown at m and n.

The glands at the end of the casing are packed with carbon, the construction of the gland being represented in Figs. 205 to 210. There are six carbon segments of the form shown in Figs. 207 and 208. These are pressed into contact with the shaft by plate

springs, as best seen to the right of Fig. 206, being carried by metal holders, to which the springs are riveted as indicated in Figs. 209 and 210. Plate springs are also used to put an axial thrust on the carbons and keep them in contact with their beds.

The reverse turbine is fitted at the after end of the casing, as indicated in Fig. 150. It consists of two stages only, each wheel carrying four rows of moving blades. The range of expansion in each stage is therefore large, and the nozzles, as shown in Figs. 181 and 182 and Figs. 183 to 185, have a considerable degree of divergence. Thus the throat section at A A (Fig. 184) is 0.675 sq. in., whilst at B B, the point of discharge, it is 1.35 sq. in. These nozzle plates are bronze castings. The nozzle openings are rectangular, and are worked out of the solid. Independent shut-off valves are provided to the end nozzles in the case of stage 1, thus making it possible to adjust to some extent the total nozzle opening to the steam supply available.

The following results were obtained on the trials of these turbines:—

Number of nozzles open	2	7	12
Revolutions per minute	301.1	479.7	593.5
Absolute steam-chest pressure ... ... lb. per sq. in.	258	249	246
Quality	0.973	0.975	0.978
Absolute pressure at first stage ... ... lb. per sq. in	19.23	49.65	86.80
Absolute pressure at sixth stage ... ... lb. per sq. in.	2.69	7.65	—
Vacuum reduced to barometer at 30 in. ... ... in.	28.26	27.96	27.3
Shaft horse-power (two turbines)	1544	6456	11,668
Water rate per horse-power per hour (all purposes, closed exhaust)... ... ... lb.	21.48	15.50	14.49
Estimated water rate (turbines only) ... ... lb.	19.98	14.2	13.45

CHAPTER XXV.

THE COMPOUND IMPULSE STEAM TURBINE.

WHILST the credit for making a practical success of the compound reaction steam turbine lies with the Hon. Sir C. A. Parsons, the honour for the introduction of the compound impulse steam turbine must be attributed to Professor Rateau. Of course, as in other cases, the idea of compounding the impulse turbine was not new at the time it was taken up by Professor Rateau; but, as every practical man knows, there is often a gulf, vast and difficult to traverse, between the claims of a patent specification and a commercial machine. Professor Rateau's title to distinction is to be found in the success with which he made this passage.

Somewhat extensive changes have been made in the construction of this turbine since its first inception, necessitated by experience gained in the practical operation of the machines. It is therefore peculiarly instructive to compare some of the earlier machines with the later, and we accordingly reproduce some engravings showing constructions, now abandoned, as well as a more complete set illustrating later practice, as embodied in the fine 5000-6000 kilowatt units built by the Westinghouse Electric and Manufacturing Company for the tramways generating station of the London County Council at Deptford.

The contract conditions here specified that the machine, when running at 750 revolutions per minute, and supplied with steam at 180 lb. gauge, and with a vacuum of 27 in., should be capable of developing in normal working 5000 kw., with a power factor of 0.85, so that the actual output demanded was about 5890 K.V.A. It was further required that each turbine should be capable of taking an overload of 25 per cent. at the same power factor, without the use of a by-pass valve. The steam consumption at full load was not to exceed 15 lb. per kilowatt-hour, with steam supplied at 180 lb. gauge and at 500

deg. Fahr., and with a vacuum of 95 per cent., or $28\frac{1}{2}$ in. Actually the builders guaranteed a consumption of 14.5 lb. per kilowatt per hour under the conditions stated, the corresponding figure at three-quarter load being $14\frac{3}{4}$ lb., at half-load 16 lb., and at quarter-load 19 lb. per kilowatt-hour.

A longitudinal section through one of these turbines is shown in Fig. 211, Plate VIII. The turbine is divided up into a series of cells or stages, each of which is, in point of fact, an independent turbine of the impulse type, and through this series of independent turbines the steam passes in succession, losing a certain fraction of its pressure head at each stage. In the present case there are twenty-four such cells or compartments, and in passing from one to the other the steam undergoes an expansion through a series of guide blades, acquiring in the process a velocity which averages about 800 ft. per second, whilst the mean blade speed is about 285 ft. per second. As will be seen from Figs. 220 to 223, each wheel has practically the same mean diameter of about 7 ft. 3 in., and, as stated, the designed number of revolutions is 750 per minute.

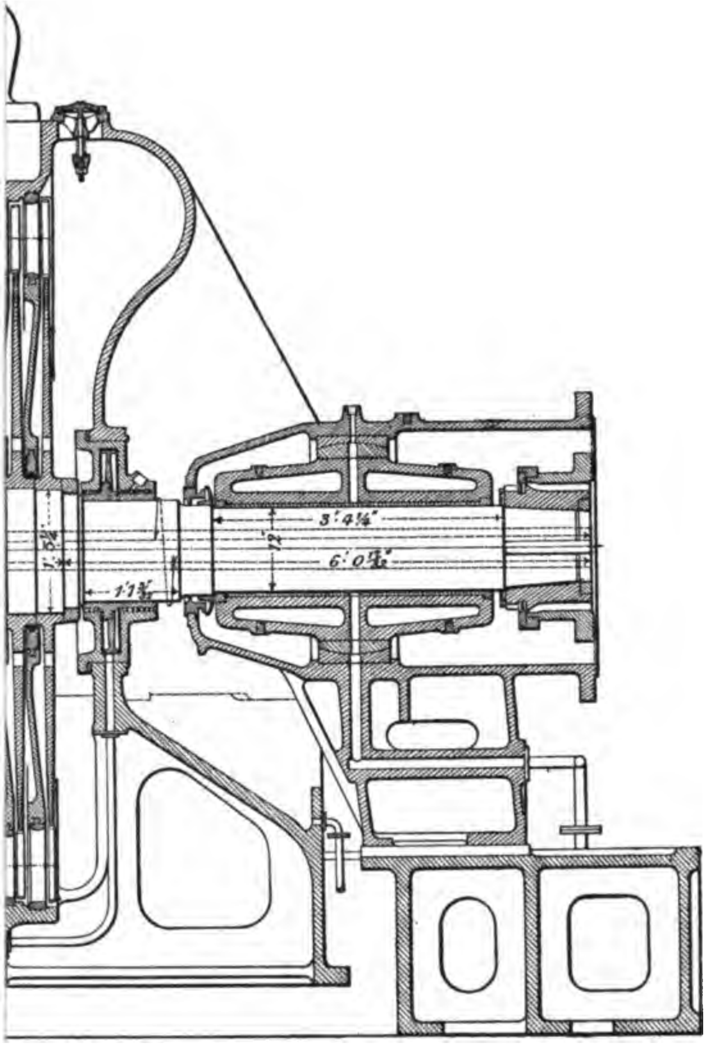
The wheels are steel forgings turned and finished all over, and assembled, by an hydraulic pressure of 20 tons, on seats turned on the turbine shaft. The latter, it will be seen, measures 1 ft. 9 in. in diameter at the centre, and is thus very stiff; a feature which is essential in turbines of the compartment-compounded type, since to keep down the leakage losses very small clearances are required between the shaft and the diaphragms.

Unless due care is taken in the design of the shaft and diaphragms, the use of these small clearances involves a certain element of risk, and it is instructive to note how any danger of injury to the machine from an accidental contact between the shaft and the diaphragm is obviated in the case of the turbine illustrated, and yet leakage losses minimised.

Each diaphragm, as shown in Figs. 224 and 225, has a dovetailed groove bored in it at the boss, and into this groove fits a keeper ring (see Fig. 226), made in halves and lined on the inside with a ring of Tandem anti-friction metal. This, in cases where the clearance permissible is small, is serrated, as indicated in Fig. 229, so that the shaft at the worst is touched merely by a

;CH POWER STATION.

[L]IMITED, ENGINEERS, MANCHESTER.



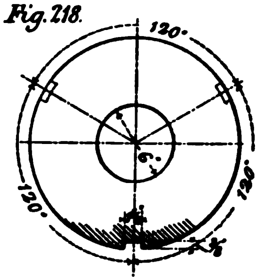


Fig. 218.

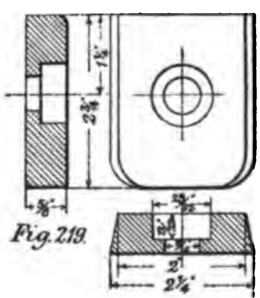
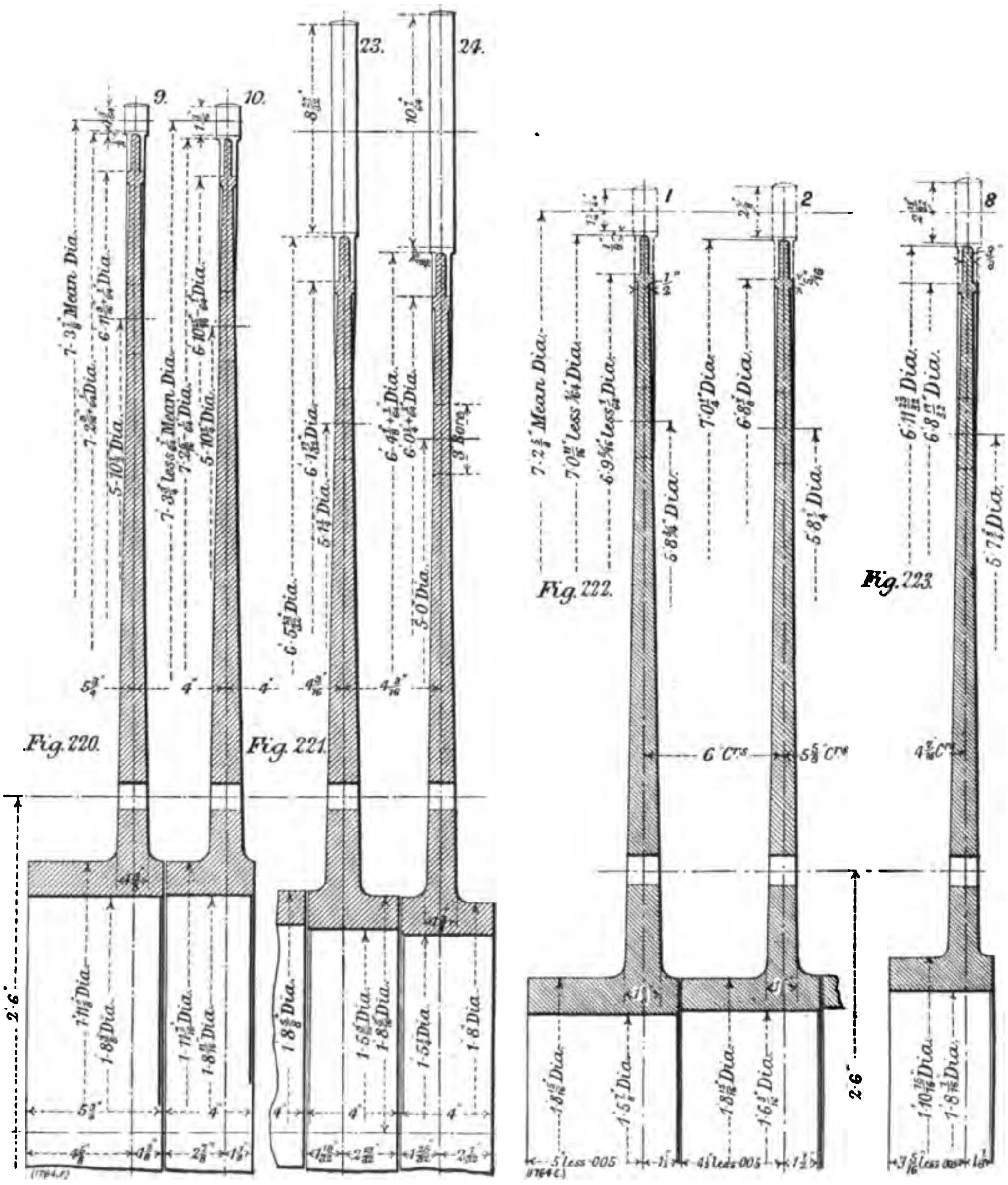


Fig. 219.

series of thin fins of a readily-fusible metal. In case of such a touch the thin fins simply yield, and no damage is done. It is therefore possible to assemble the turbine with no initial diaphragm clearances



Figs. 220 to 223. Wheels for Rateau Turbine.

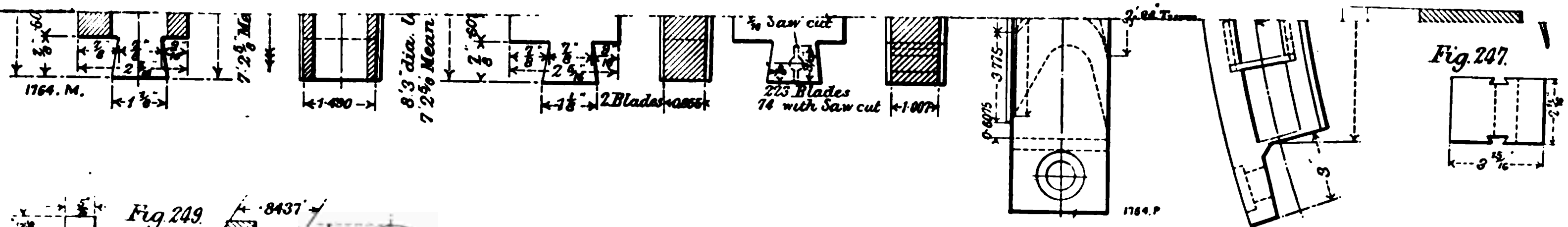
at all in the case of the first nine diaphragms, the tandem metal rings being scraped to fit the shaft. For the remaining diaphragms a positive clearance of a few mils is allowed, and the tandem metal

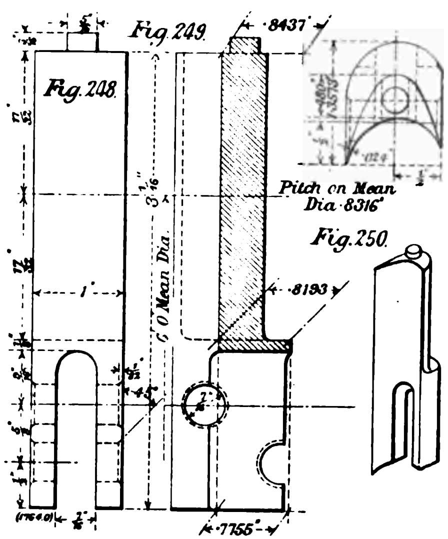
is then turned in a series of grooves, as indicated in Fig. 232, Plate IX. The carrier-rings are prevented from turning in the diaphragms by cheese-headed bolts, the heads fitting partly in the diaphragm and partly in the carriers, as indicated in Figs. 225, 227, and 228, where, in the last-named figure, the arc of the circle shown on the dovetail indicates the spot bored out to receive the head of a $\frac{3}{8}$-in. screw tapped into the casting, as indicated in Fig. 225.

The diaphragms are steel castings slightly dished, as indicated in Fig. 224, Plate IX., which represents Diaphragm No. 24, the last of the series. This diaphragm is bladed all round, as are the whole of the diaphragms, for the low-pressure end of the turbine. They are all made in halves, as shown, being cast with lugs (shown dotted in Fig. 224) to facilitate machining. These are ultimately machined off. At the horizontal joint a groove to take a key is cut in each half of the diaphragm, being undercut in the case of the lower half. A key is driven into this groove, and secured in place into the lower half diaphragm by countersunk screws. This key thus forms a tongue which fits into the groove in the upper half of the diaphragm, rendering the joint practically steam-tight.

Details of the guide blades for this diaphragm are given in Figs. 234 to 242. These blades are malleable castings, which on the convex faces are filed up to gauges, and are then registered and clamped into a former. With a group of blades thus assembled, the dovetails at the bottom are cut in a vertical milling machine, the table of which is constrained to follow a guide so as to give the proper curve to the tails, which have, of course, to fit on to a circular, and not a straight surface. The concave face of the blade is finished by milling. One of the blades, Figs. 234 to 236, is extra stout, and is drilled to take a $1\frac{1}{8}$-in. bolt, which (see Fig. 231) passes through the centre of the upper half of the turbine casing, through the blade and into the diaphragm, thus ensuring its being lifted when the top cover is removed. The "closure" blades, which come on each side of the horizontal joint, have the shape shown in Figs. 237, 238, and 239; but the remainder are made as in Figs. 240, 241, and 242. The hole at the top, indicated by the dotted lines, is drilled in all blades near the joint, as indicated in Fig. 233. These holes take pegs fixed in a band 1 in. by $\frac{3}{8}$ in. in section, which passes round the outer ends of the blades lying in the groove shown. Every

DETAILS OF 5000-KILOWATT RATEAU STEAM TURBINE AT GREENWICH.





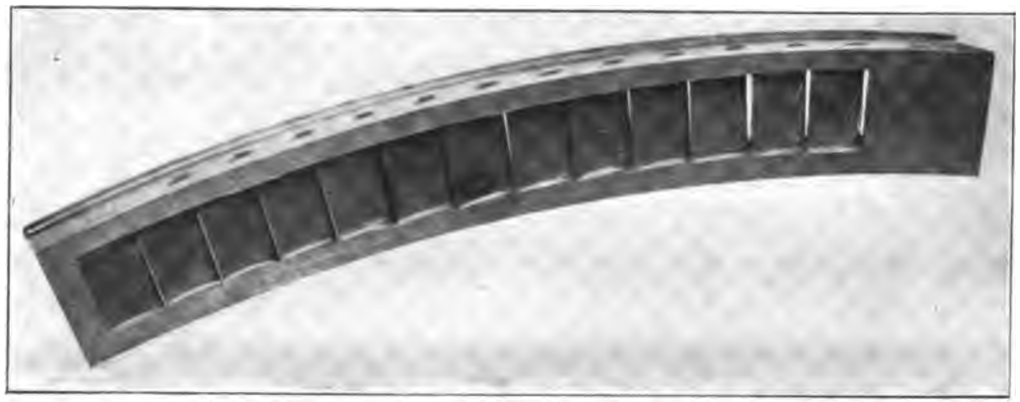
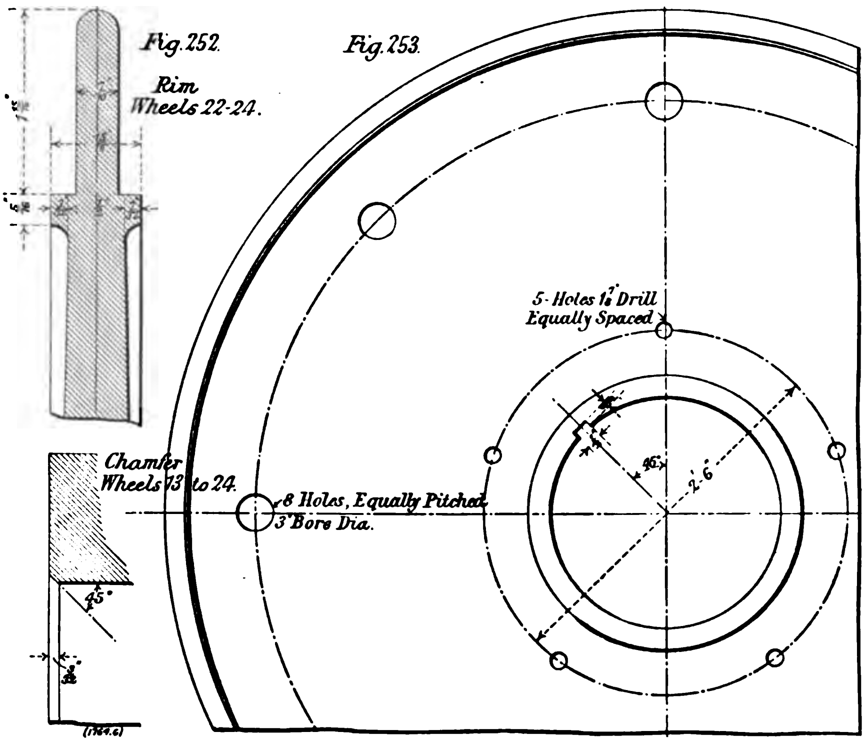


Fig. 251. Set of High-Pressure Guide Blades.

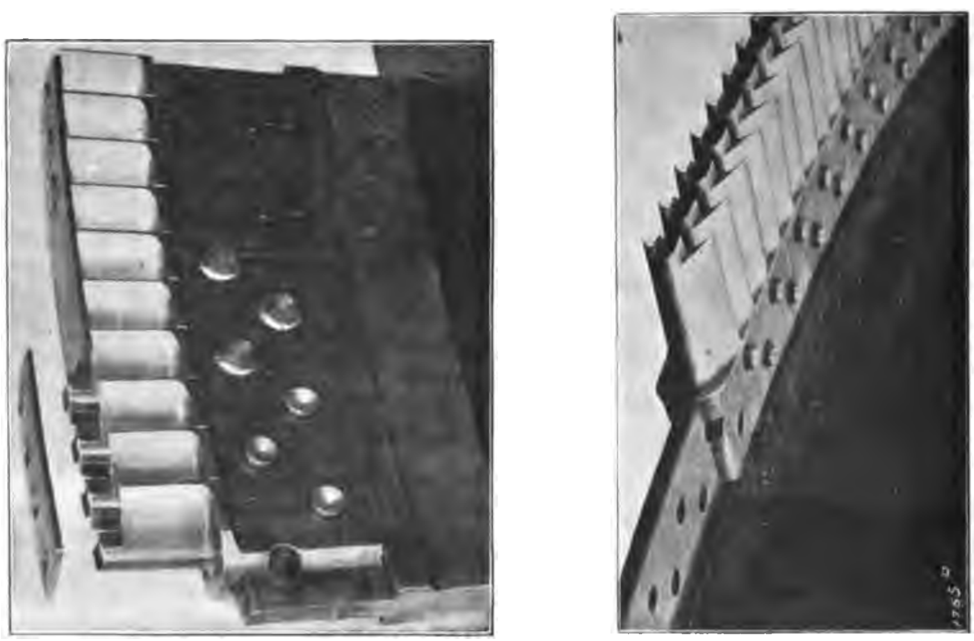
third blade, moreover, has its dovetail at the root split by a saw cut and drilled as shown at the lower end of Fig. 241. In assembling the blades a taper peg is driven into the drilled hole, which thus wedges the blade firmly in its groove. The "closure" blades are held in place by stopper plates secured by countersunk set screws, as indicated in Fig. 230. The discharge angle of these low-pressure blades lies between 26 deg. and 30 deg., this relatively large value being adopted so as to secure the requisite area of steam way without the use of abnormally long blades.

At the high-pressure end of the turbine the admission is partial, and the arc of circumference subtended is here increased by reducing the discharge angle of the blades to a value lying between 13 deg. and 16 deg. The blades in this case have the form indicated in Figs. 245, 246, and 247, which represent those used for Diaphragm No. 2. These blades are cut out of rectangular steel bars, which are first drilled to a diameter corresponding to the radius of the concave face of the blades. Two blades are secured from each bar, and the convex and straight surfaces are finished by milling. The finished blades are tinned at the ends, and cast into a segment piece, as shown in Figs. 243 and 244. A complete segment is shown in Fig. 251. It fits into recesses machined for it in the diaphragm. Some builders of Rateau turbines, we may add, use drawn-steel sections for the guide blades, which are cut to length and cast in. The position of the blades in each successive diaphragm is so situated with respect to those in the preceding diaphragm as to utilise as fully as possible the residual kinetic energy in the steam as it escapes from the moving blades; and for the same reason a small "entrance" angle (32 deg. to 40 deg.) is in this case given to the guide blades, as illustrated in Figs. 246 and 247. After each diaphragm is finished the blades are filled in with lead, and the strength of the diaphragm tested by water pressure.

Details of several of the moving wheels are illustrated in Figs. 220 to 222, page 241, and in Figs. 252 and 253, page 244. The blades vary considerably in dimensions, but are all constructed on the same system, having the form shown in Figs. 248, 249, and 250, Plate IX. They are forked at the roots, and straddle the discs (see Figs. 254 and 255, page 244), being secured by rivets. These rivets are a driving fit for their holes, and are closed by rolling, and not by



Figs. 252 and 253. Wheel for Rateau Turbine.



Figs. 254 and 255. Blades for Rateau Turbine.

hammering, as the latter operation might possibly distort the disc. When the disc is finished the rivet heads and projecting ends of the snugs are ground off, so that the wheel, when ready for erecting, has a perfectly smooth finish. The wheels are assembled on their seats by hydraulic pressure, but two $1\frac{1}{2}$-in. by $\frac{3}{4}$-in. keys are also provided, though only one is shown in Fig. 253. Any danger of the wheels shifting axially when in place is obviated by a lock nut at each end (see Figs. 211 to 214, Plate VIII). These nuts are prevented from unscrewing by a stop, shown in position in Fig. 214, and separately in Fig. 219. Each wheel seat or step on the shaft is tapered at the end, as indicated in Fig. 217, so as to facilitate the putting of the discs in place. Each of the discs is drilled with five $1\frac{7}{8}$-in. tap holes, into which rods can be screwed, should it be necessary to remove a disc. There are also eight larger holes (3 in. in diameter) round the circumference, to ensure an equality of steam pressure on the two sides of the disc.

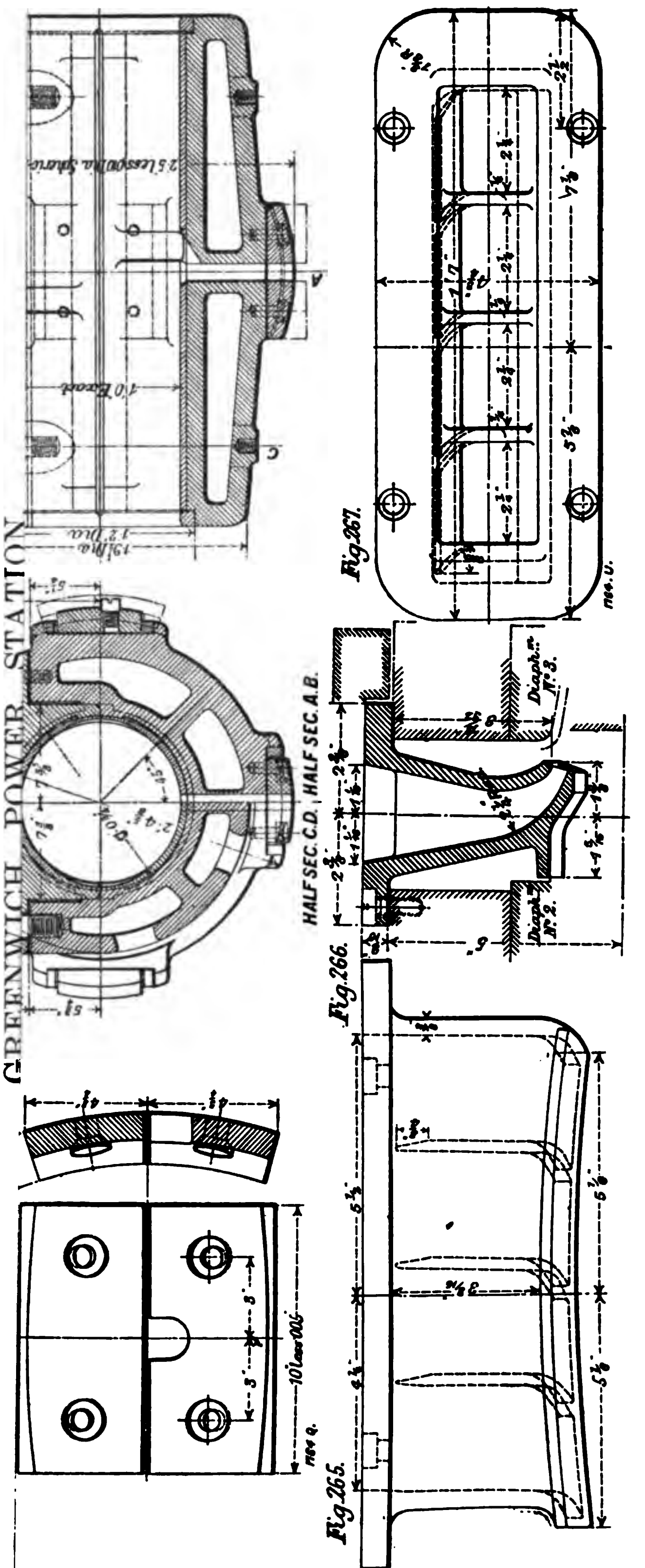
Each wheel, after completion, is balanced, first statically, and is then mounted on the shaft, which, after the addition of each successive wheel, is balanced dynamically.

The blades are cut out of solid 5 per cent. nickel-steel bars. The blanks are rectangular, and the first operation consists in rough-milling these with two parallel sides to fit a gauge. A gang mill is used which finishes both sides at once. A number of the blanks are next assembled in a jig, and the forks are then milled by a five-gang cutter. The group is next turned end for end, and the snugs, to receive the shrouding, milled with a three-gang cutter. This operation makes the blades true to length, and the concave faces are now milled. The mill used has inserted cutters of high-speed steel, but a fairly low rate of speed and feed is used in order to preserve the cutters. After this operation each blade is clamped (with its concave face) to a hardened mandrel, and turned down in a lathe to the thickness required. A hardened shoulder on the mandrel serves as a gauge to which to set the tool, so that calipering is dispensed with. This operation leaves the blade with its front and back faces concentric, though of the right thickness at the middle of the blade. The next operation is to thin the blade down at the entrance and exit by removing the superfluous metal at the back, thus giving the section indicated in Fig. 250.

This operation is effected in a profiling machine in the case of short blades, but in the case of long blades two operations are required. A formed cutter removes all the superfluous metal, save near the shoulder, this being left for a finishing operation in the profiling machine. After this operation the blade is nearly finished, but the root is still rectangular, and it is therefore machined at the back, so as to fit the concave face of the adjoining blade. In this operation the blade is also tapered so that the back is truly radial to the disc on which it is to be mounted. This operation concluded, the blades are assembled in a jig in pairs in order to have the rivet holes in the forks drilled.

An end view of the turbine, showing the governor and the position of the main valve box, is represented in Fig. 256, Plate X. From the valve box the steam is led by a curved pipe to two nozzle boxes situated diametrically opposite each other. These nozzle boxes are of a somewhat special construction (see Figs. 257 to 259). They have, it will be seen, adjustable openings, the steam-way being capable of being increased or diminished by turning the hand wheel shown, which, by means of gearing, raises or lowers simultaneously the three spindles, each of which carries at its lower end a block scraped to fit between the parallel faces of the nozzle segment. By lowering these blocks the steam way can be diminished, and the output of the turbine reduced without throttling the supply at the governor valve. Hence for a very large range of load the first wheel can be worked with constant efficiency. This arrangement was particularly serviceable in the present instance, as the County Council specification contained a requirement to the effect that the turbine should take a 25 per cent. overload without the use of a by-pass valve, the latter only being used when the demand on the turbine exceeds 6250 kilowatts. A section through the main valve box is represented in Fig. 260. The steam stop valve is of the balanced type, and directly below it is fitted the emergency cut-off valve, which comes into operation if the turbine begins to race. From this valve the steam passes through a strainer to the underside of the ordinary governor valves, of which there are two, as shown, this plan being adopted with a view to obtaining a maximum steam way, with a minimum of weight to be shifted by the governor. The automatic overload governor valve is shown to

DETAILS OF 5000-KILOWATT RATEAU STEAM TURBINE AT THE GREENWICH POWER STATION



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the right, and admits steam through a special nozzle box to the third stage. The non-condensing by-pass is hand-operated, and is shown to the right of the overload valve, and admits steam through a similar nozzle box to the overload one to the fifth stage. These special nozzles are shown in Figs. 265 to 267, Plate X.

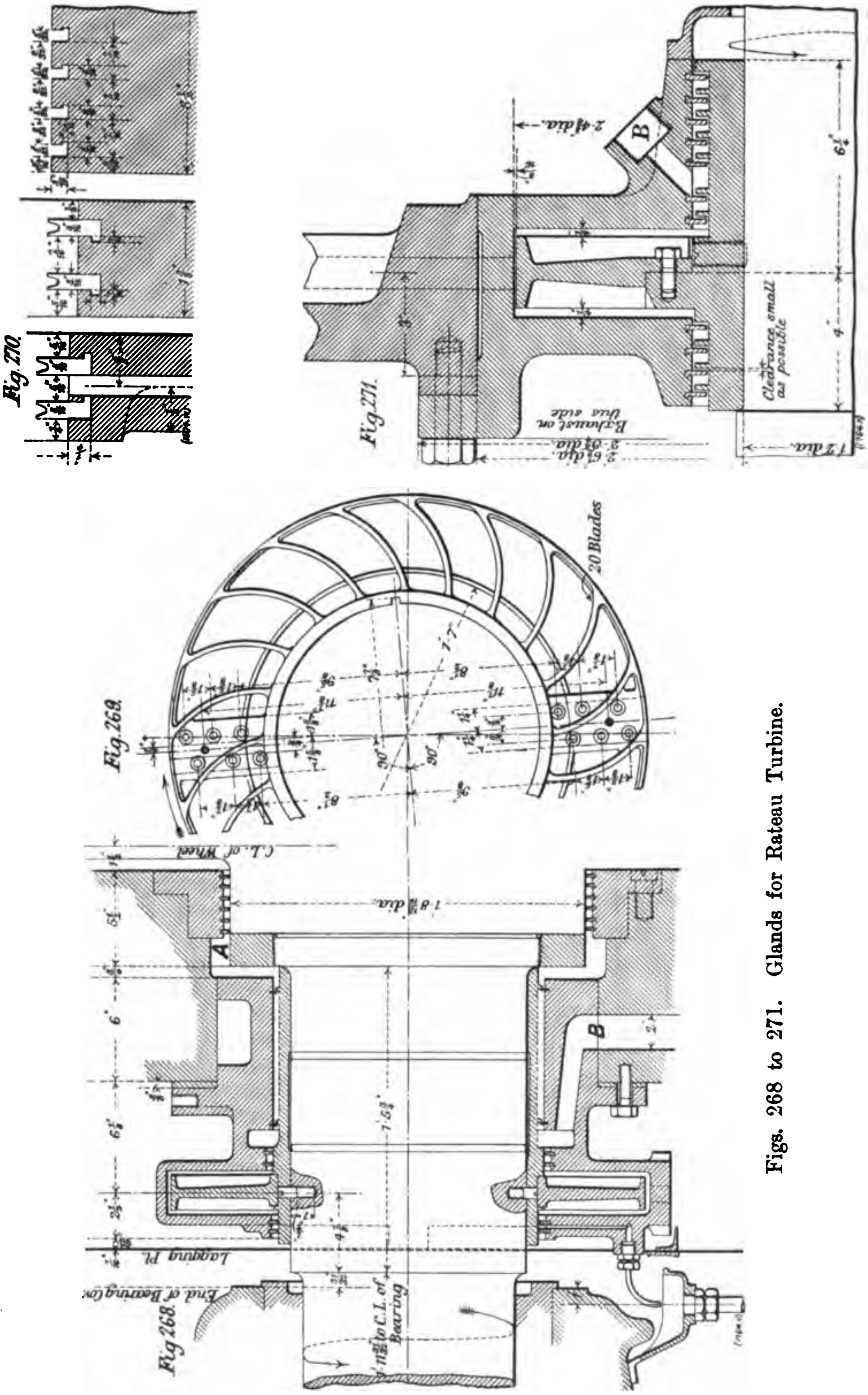
Compartment-compounded impulse turbines have generally fixed bearings, as these tend to stiffen the shaft. In the present case, however, the specification called for spherical-seated bearings, and details of those fitted are given in Figs. 261 to 264. The shells are lined with white metal, and the oil supply is pumped in through $1\frac{1}{4}$-in. holes passing through the lower pads. The side pads are split, as shown in Fig. 264.

The turbine glands, as shown in Figs. 268 to 271, page 248, are of the water-sealed type. The high-pressure gland is shown in Figs. 268 and 269. The steam pressure to which the gland is exposed is about 100 lb. absolute, and hence careful packing is required if leakage is to be reduced. To this end labyrinth packing is provided comprising two sets of dummies. The first set are of the radial-flow type, and reduce the pressure to about 20 lb. absolute, the leakage from chamber A being returned to a stage in the turbine with a corresponding pressure.

The second set of dummies are of the axial-flow type, and reduce the pressure to that of the exhaust, chamber B being connected to vacuum. The paddle wheel is thus working under similar conditions to the low-pressure one. The construction of the paddle wheel is shown in Fig. 269.

As is well known, impulse turbines are almost free from end thrust, but a thrust block has nevertheless to be fitted in order to maintain correct alignment axially. This is shown to the left of the bearing in Fig. 211, Plate VIII. Details of the grooves on the shaft into which the thrust collars fit are given in Fig. 216.

The spiral gears, by which the governor and oil pump is driven, are arranged to the left of the thrust block, and to the left of this, again, comes the emergency governor. This is simply a round-headed bolt passing axially through the shaft and held down to a seat by a spring. The head of the bolt being heavier than the shank, tends to fly out under the centrifugal force developed by the rotation of the shaft. Should the turbine exceed the speed limit



Figs. 268 to 271. Glands for Rateau Turbine.

fixed, the centrifugal force in question becomes sufficient to overbalance the spring, and the head of the bolt flying out trips a detent,

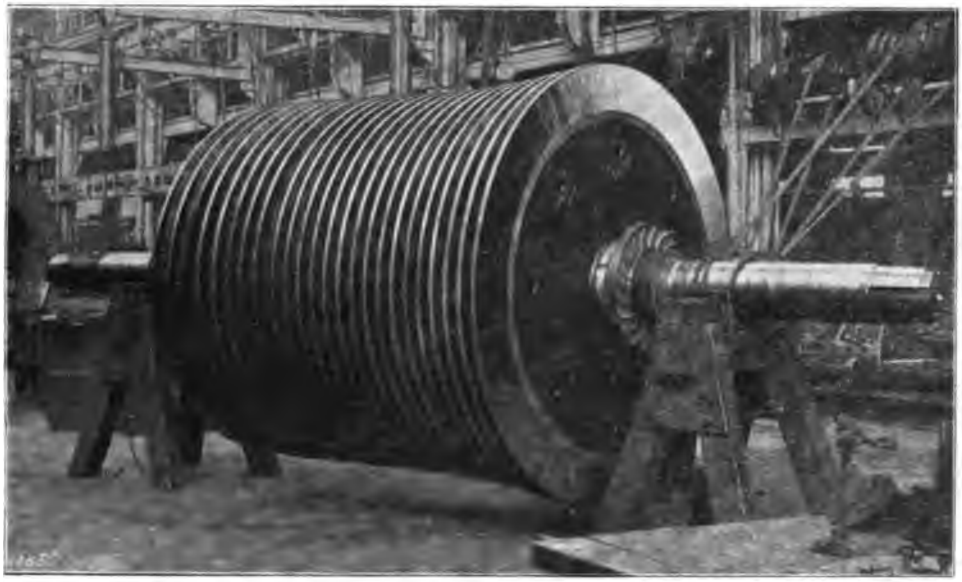


Fig 272. Rotor of Rateau Turbine.

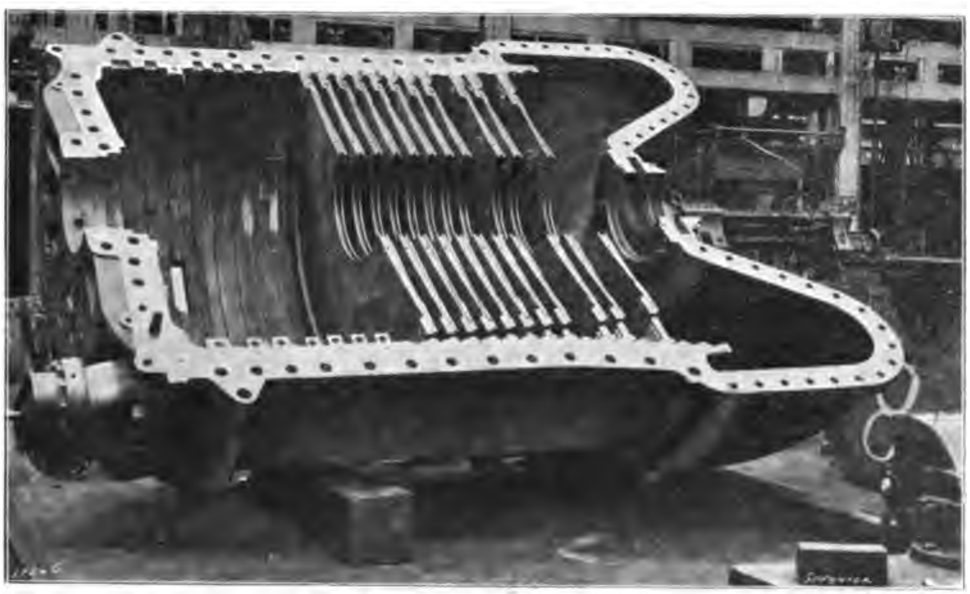


Fig. 273. Upper Half of Casing ; Rateau Turbine.

and the emergency cut-off valve then flies on to its seat. A view of the rotor complete is given in Fig. 272, and of the upper half of the casing in Fig. 273.

The weight of each turbine complete is 85 tons, of which total the rotor accounts for 25 tons.

The condenser supplied has 12,500 sq. ft. of cooling surface, and is provided with a Leblanc air pump, which on trial has easily maintained a vacuum of $28\frac{1}{2}$ in., the ratio of cooling-water supply to the steam condensed being 65 to 1.

In Figs. 274 to 276, Plate XI., the general arrangement of a 600-kw. turbo-generator of an earlier pattern is illustrated, as constructed for a speed of 3000 revolutions per minute, by Messrs. Fraser and Chalmers, Limited, of Erith. A general view of it is also given in Fig. 278, Plate XII., and a half-section through this turbine is represented in Fig. 279.

As originally made, the wheels of Rateau turbines were flanged plates, the buckets being riveted on to the flanged rim. This construction, though unobjectionable at moderate bucket velocities, proved unsuitable for high speeds, as there was a tendency for the flange to bend outwards under the centrifugal forces. The arrangement next adopted in the Rateau turbine was as shown in Fig. 279, and on an enlarged scale in Fig. 280, annexed. Each wheel, it will be seen, was made up of two discs of nickel steel, each $\frac{3}{16}$ in. thick. These were riveted to the flange of a central hub of mild steel, keyed to the shaft, and the discs were also riveted together near their peripheries. A quarter view of the hub is shown separately in Fig. 281, Plate XII.

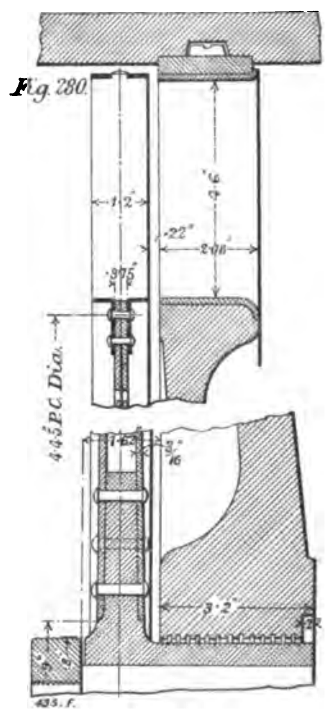


Fig. 280. Diaphragm Details.

The blades were stamped out of high-quality nickel-steel strip, 0.06 in. or 0.08 in. thick, to the forms shown in Figs. 282 and 283, page 251; or, perhaps still better, in the perspective view, Fig. 284, where also will be seen the blank before it is stamped to shape. As will appear from these figures, the blades were forked at the lower end, and when mounted in place these forks straddle the discs, and the blade was then secured in place by two $\frac{5}{32}$-in. rivets,

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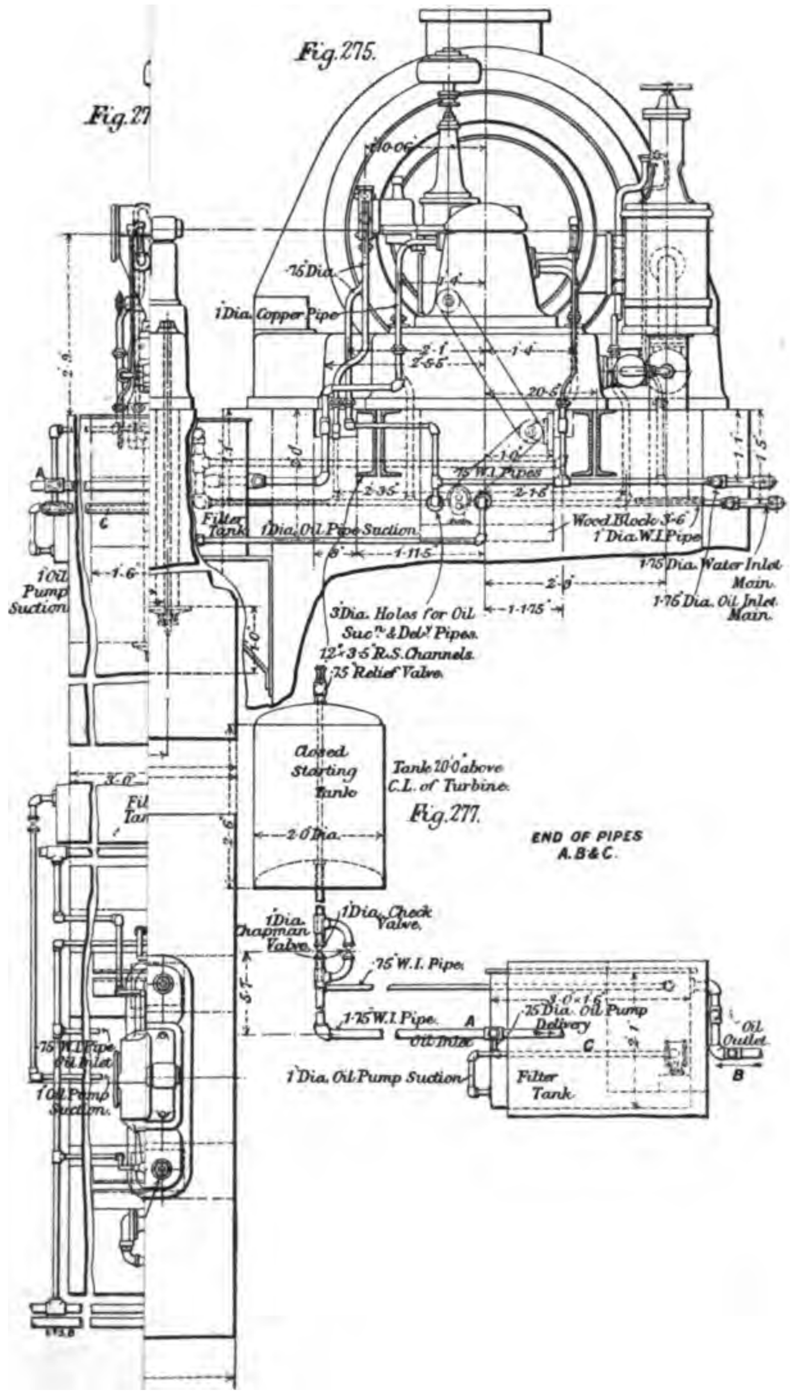


Fig. 275.
Fig. 27
.75 Dia.
1 Dia. Copper Pipe
.75 W.I. Pipes
Filter Tank
1 Dia. Oil Pipe Suction
Wood Block 3-6
1 Dia. W.I. Pipe
1.75 Dia. Water Inlet Main.
1.75 Dia. Oil Inlet Main.
1 Oil Pump Suction
3 Dia. Holes for Oil Suc.n & Del.y Pipes.
12" × 3.5 R.S. Channels.
.75 Relief Valve.
Closed Starting Tank
Tank 10-0 above C.L. of Turbine.
Fig. 277.
END OF PIPES A.B&C.
2-0 Dia.
1 Dia. Chapman Valve.
1 Dia. Check Valve.
.75 W.I. Pipe.
1.75 W.I. Pipe.
.75 Dia. Oil Pump Delivery
Oil Inlet
Oil Outlet
1 Dia. Oil Pump Suction
Filter Tank
.75 W.I. Pipe Oil Inlet
1 Oil Pump Suction.

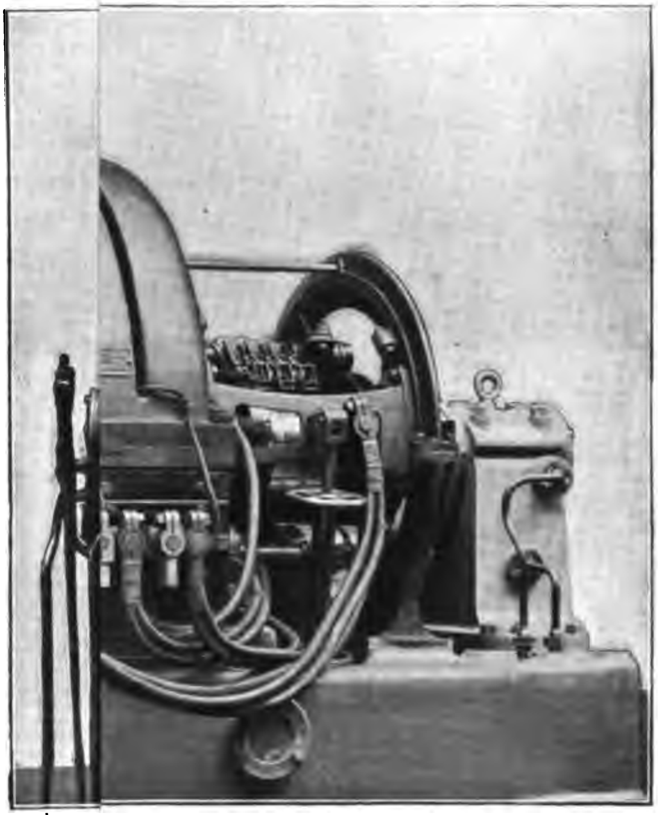
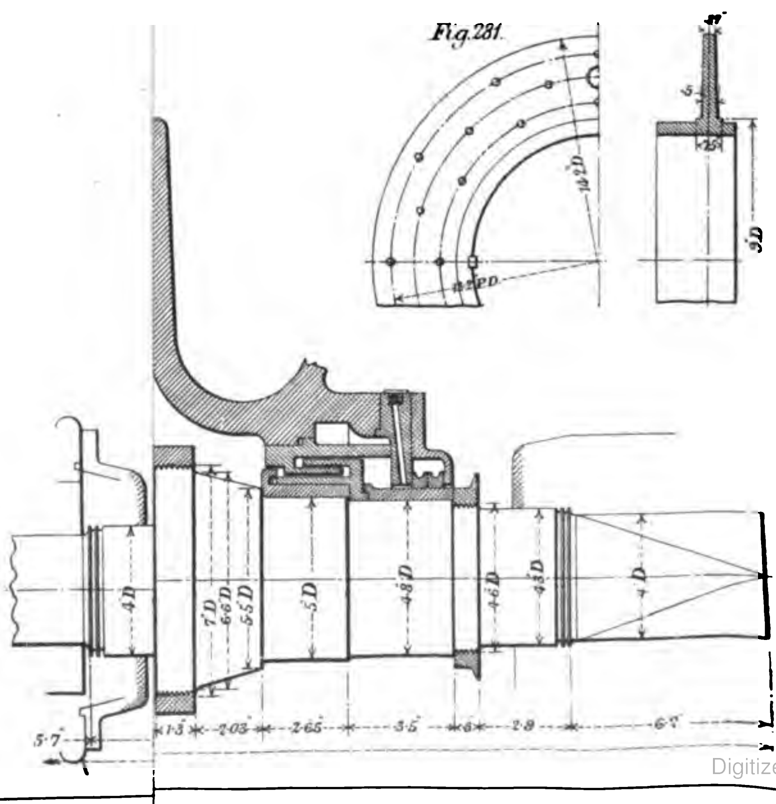


Fig. 281.



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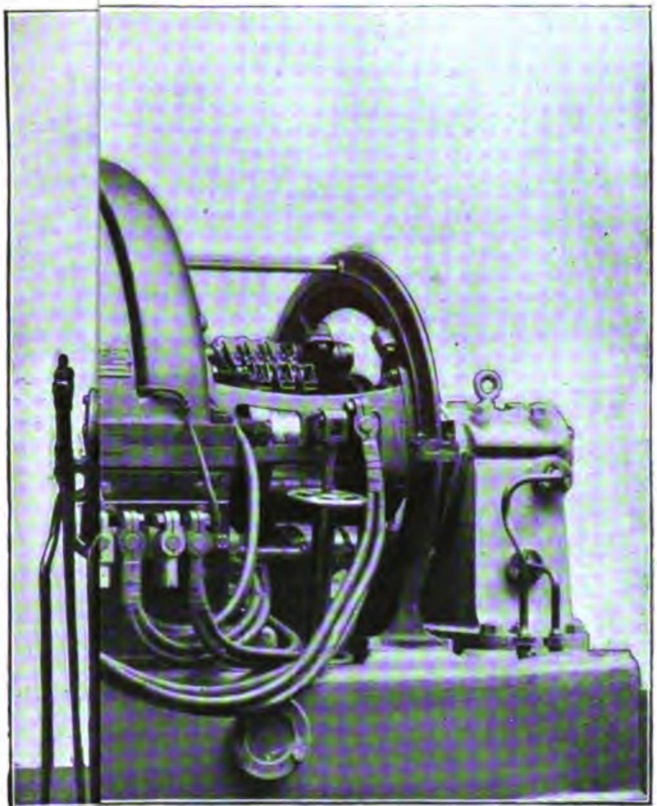


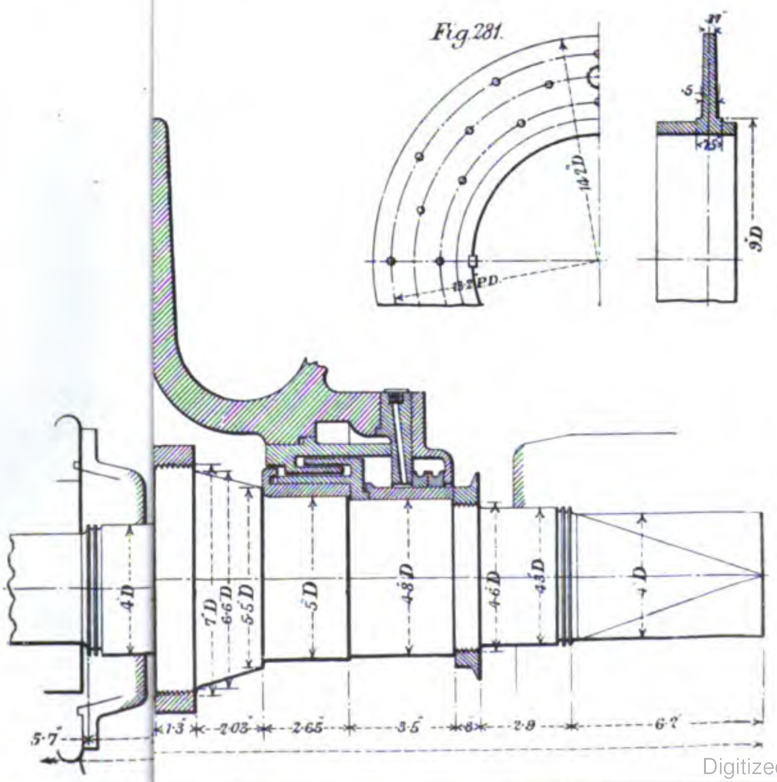
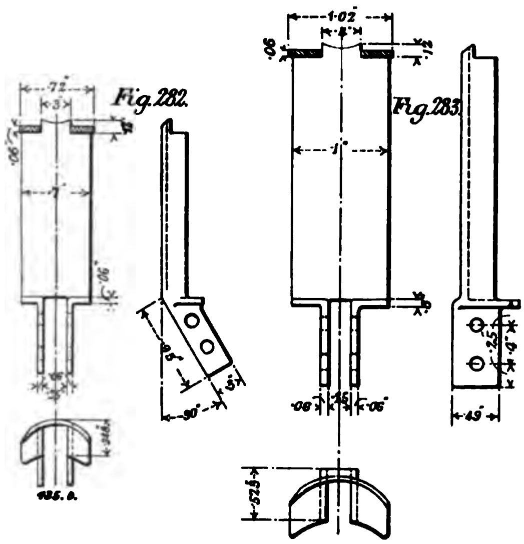


Fig. 281.
14·2D
9D
18·2 P.D.
·5
·25
4D
7D
6·6D
5·5D
5D
4·8D
4·6D
4·5D
4D
5·7
1·3
2·03
2·65
3·5
·8
2·9
6·7

To Face Pa

as shown in Fig. 280. This method of mounting the blades is still adhered to in the case of Rateau turbines and makes a very solid job, but the plan of stamping the blades out of sheet nickel steel has been abandoned, and they are now machined out of the solid. In fact these sheet-steel blades have a considerable amount of trouble from stripping. It was not found possible to secure sheets of 32 per cent. nickel steel of uniform quality throughout, and 25 per cent. nickel steel here, as in other cases, proved an utterly unreliable material. Even blades made from sheets of 5 per cent. nickel steel could not be relied upon to give satisfactory service,



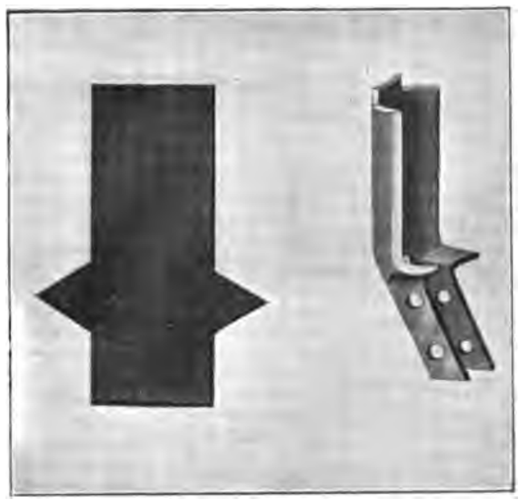


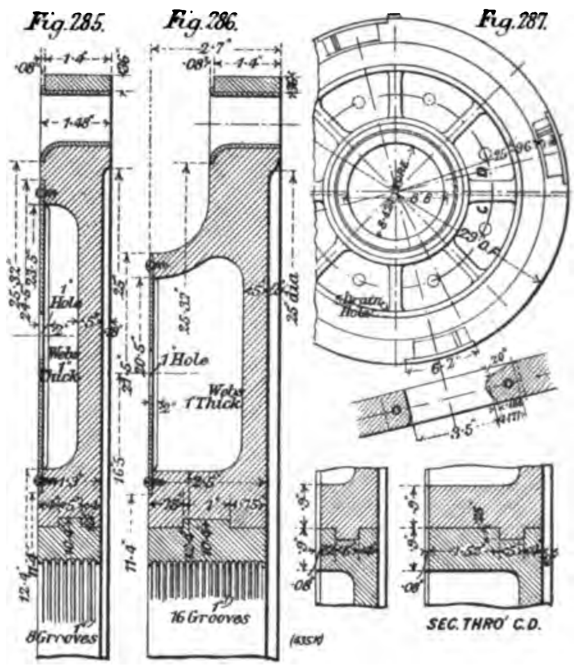
Fig. 284.

Figs. 282 to 284. Blades (Old Type) for Rateau Turbine.

and it was therefore decided to cut them out of the solid. With blades thus prepared by machining it is possible to put up the blade speeds, since, where the centrifugal stresses render it desirable, as in the case of long blades, they can be thickened at the roots. Owing to the higher speeds permissible, fewer of these machine-cut blades are required.

The construction of the diaphragms in this turbine is as illustrated in Figs. 285 to 287, page 525. The guide blades were bent out of sheet brass and riveted up into sets, as indicated in Fig. 288. These sets of assembled guide blades were then fitted into the diaphragms, as shown in Figs. 285 and 286, and also in a large scale on the right of Fig. 280.

A section through an exhaust turbine of 1000 horse-power is shown in Fig. 289, on the opposite page. It was designed for 1500 revolutions per minute, and showed a brake efficiency ratio of 66 per cent.



Figs. 285 to 287. Diaphragm Details.


Fig. 288. Set of Guide Blades.

A pattern of the compartment-compounded turbine, differing in some mechanical details from that developed by Professor Rateau, has been introduced under the name of the Zoelly turbine, and Figs. 290 and 291, Plate XIII., refer to a Zoelly turbine and generator developing 850 kilowatts at 1500 revolutions per minute, built by Messrs. Mather and Platt for the Risca Colliery of the United National Collieries Company in South Wales. A longitudinal section through the turbine is shown in Fig. 292, Plate XIV., and an enlarged view of the low-pressure gland is given separately, which will be referred to later. There are, as will be seen, sixteen wheels on the shaft, each in its own compartment. The mean diameter of the smaller wheels is 53 in. and of the larger 62.4 in. Steam enters the annular steam chest on the extreme left of the casing and passes thence to the first wheel chamber through nozzles cored in the circular casting shown. After doing its work on the wheel, the steam enters the next chamber through similar nozzles formed in the diaphragm between the chambers, and gives up the velocity acquired in these nozzles to

the second wheel. Thence alternately through nozzles and wheel blades it passes away to the exhaust at the right-hand end of the casing. The nozzles are formed by casting curved iron dividing plates, about $\frac{1}{8}$ in. thick in channels cored in the diaphragm castings. The dividing plates are spaced at about 2-in. pitch, and direct the steam at an angle of 20 deg. to the plane of the wheel. The nozzles at the high-pressure end are about $\frac{3}{8}$ in. deep radially, and are disposed in two groups on opposite ends of a diameter. The depth

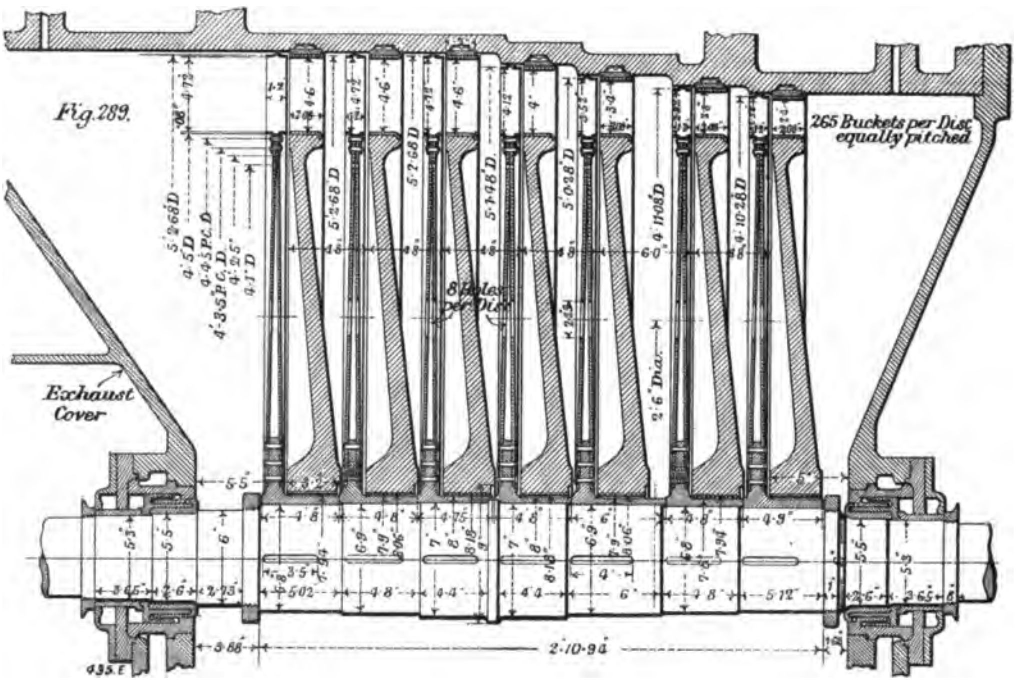


Fig. 289. 1000-Horse-Power Rateau Exhaust Turbine.

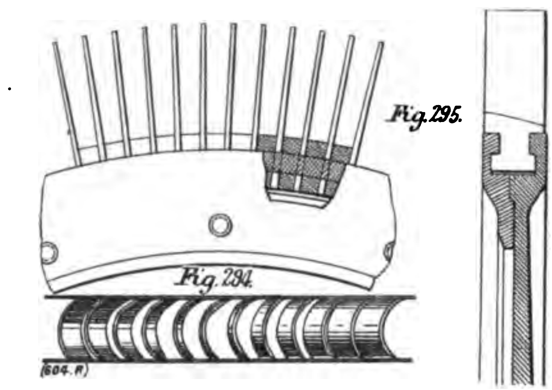
is maintained constant for as long as possible, the extra steam way necessitated by the greater volume of the steam in successive compartments being obtained by adding extra nozzles to each group. Finally, when all the available circumference is filled by nozzles, the latter are increased in depth to accommodate the increasing volume of the steam.

The diaphragms are of cast iron, slightly dished, and with horizontal flanges along the joint, through which temporary bolts are put to hold the halves together while being machined. Nozzles are cast in as already mentioned, and the diaphragms are located longitudinally by circumferential flanges, which form spacing pieces

between them. Set screws through the casing hold the half-diaphragms from moving when the cover of the turbine is lifted. The central hole in the diaphragms is bushed with bronze, the bore of the bushing being turned with numerous grooves. The wheel boss runs inside the bushing, with a clearance all round of about 1 millimetre.

The turbine wheels are made from about 5 per cent. nickel steel, and are very highly finished all over, to minimise disc friction. The Zoelly wheels vary from about 2 centimetres in thickness at the boss to about 1 centimetre under the rim. About half-way

Fig. 293.



Figs. 293 to 295. Blading of Zoelly Turbines.

between the boss and the rim the wheel is thickened up by a facing on each side, and through this thickened part half-a-dozen large holes are drilled. The function of these holes is to permit of the instantaneous equalisation of the pressure on each side of a wheel when the load is suddenly thrown on or off. Any end thrust there may be is taken by a simple thrust block, the object of which, however, is more to locate the rotating parts endwise than to do anything else.

The blades of the Zoelly turbine are of 5 per cent. nickel steel, and photographs of them, together with the distance pieces which go between them, are reproduced in Fig. 293. Figs. 294 and 295 indicate the way in which they are held in position.

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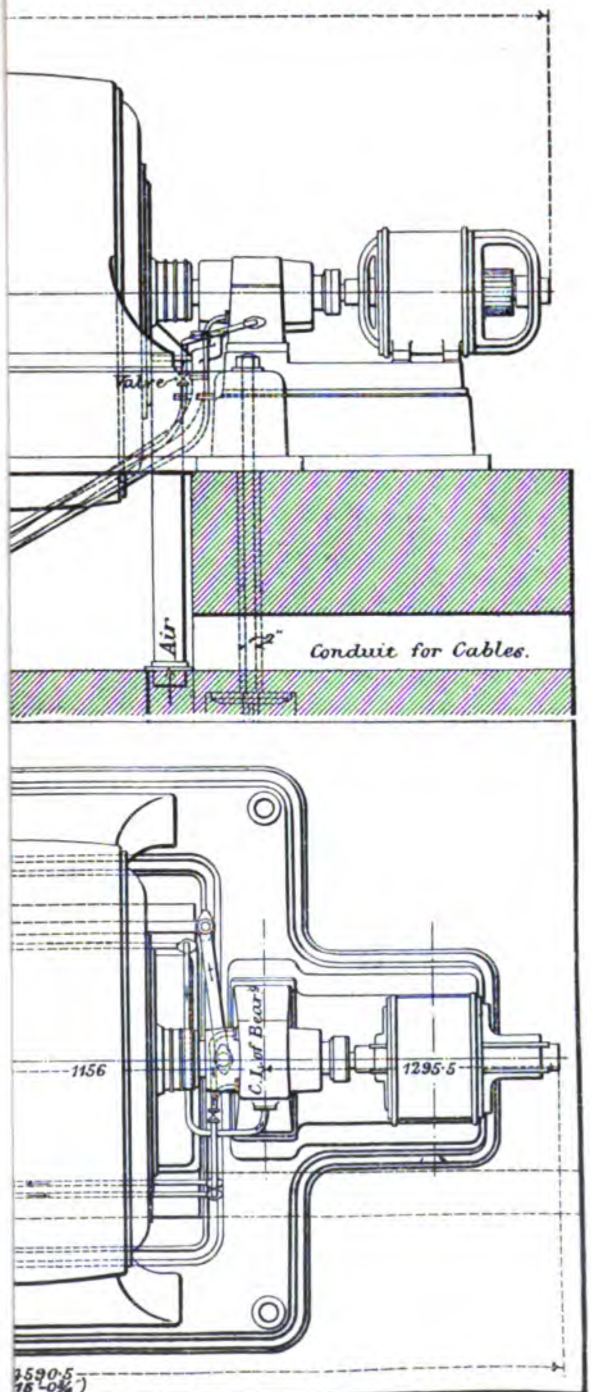

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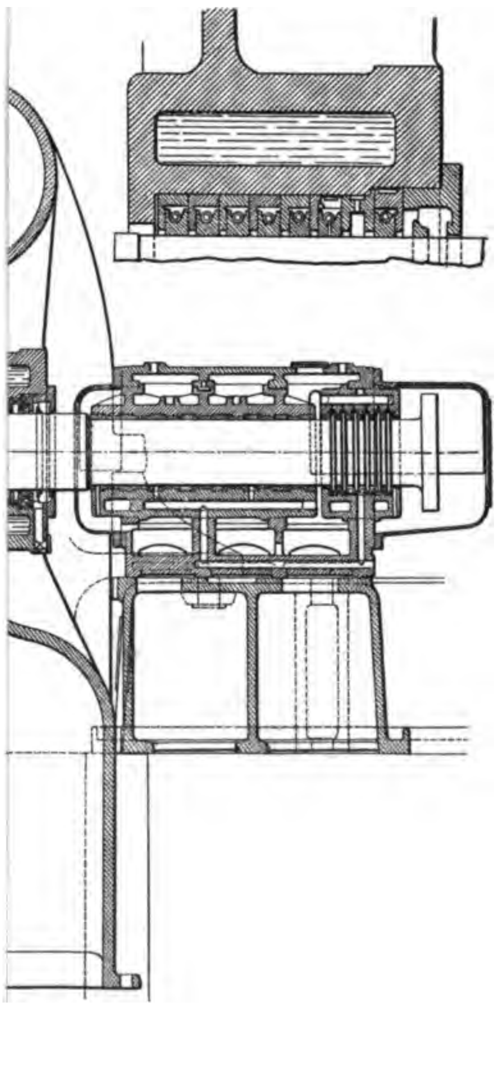

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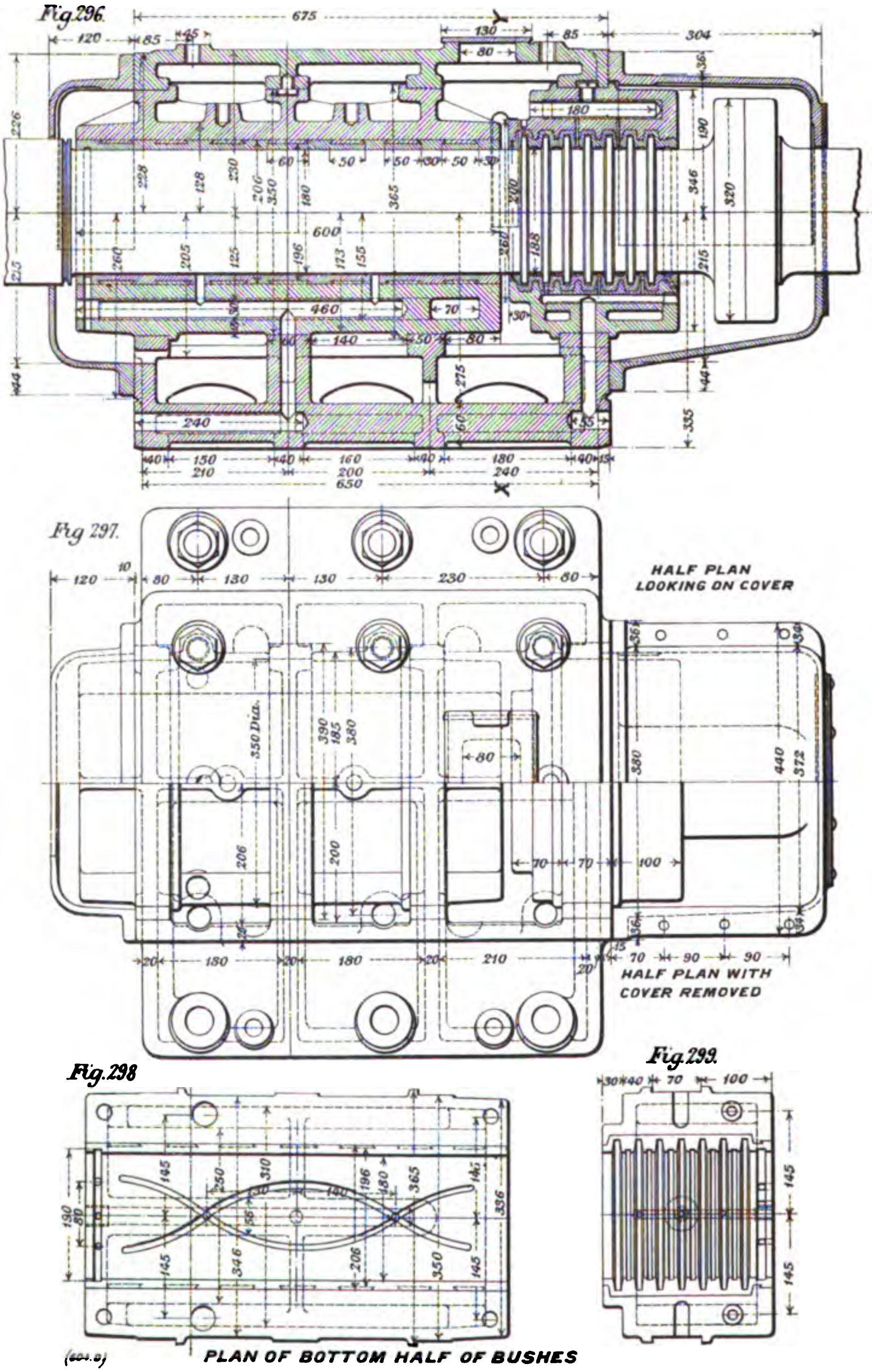
They are made of sheet metal, bent to the curvature required, and notched at the root in the manner shown. They are ground to a good finish, both internally and externally, and the back of the leading edge is also ground away so as to obtain a very fine edge. Their thickness is about a millimetre, but the longer blades taper down in thickness from root to tip. The distance pieces are also of steel, machined all over, and polished on the upper face, which is in contact with the flow of steam.

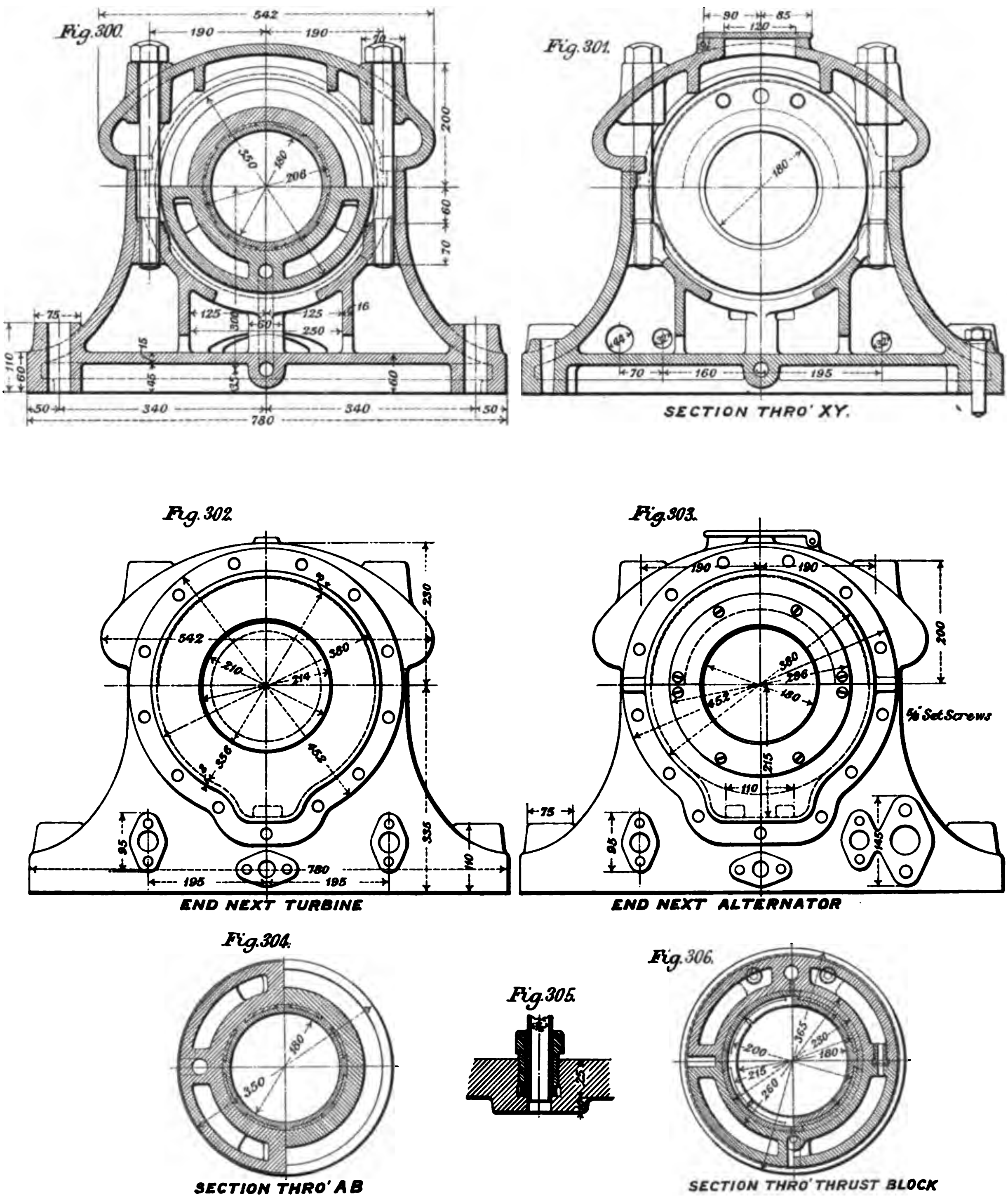
Fig. 295 indicates very clearly the manner in which the blades are held. They are laid in position alternately with the distance pieces when the wheel is on its side, a length of square iron rod curved to the correct radius being cramped to the wheel to locate the notches on the then upper side. When the whole of the blades are inserted, the temporary locking iron is removed, and replaced by a ring which secures all the blades in position. This ring is then riveted to the wheel, as shown. In recent designs, we may add, the blades have been shrouded at their outer edges. They run with about 3 millimetres clearance from the nozzles, and have about the same clearance between their tips and the casing of the machine.

The necks of the bearings and the portions of the shaft running in the glands are lapped to the highest possible finish. The central part upon which the wheels are pressed is made in five successive diameters, each 1 millimetre greater than the previous one. Each diameter carries a group of three wheels, except the end one, which carries four. The wheels are secured longitudinally between a fixed collar at one end and a screwed collar at the other, the boss of each wheel being in contact with those on each side, so that by tightening up the collar the whole series is locked. The collar is then itself fastened by a steady pin. Feathers let into the shaft, and secured by countersunk screws, prevent the wheels from turning relatively to the shaft. There is one feather for each group of wheels on the same diameter of shaft.

The bearings are fixed to the bed plate, and are entirely distinct from the turbine casing, so that they receive no heat from the latter. Details of the bearing at the low-pressure end are given in Figs. 296 to 306, pages 256 and 257. Both bearings are water-cooled and provided with forced lubrication. The steps are lined with white



Figs. 296 to 299. Main Bearings; Zoelly Turbine.



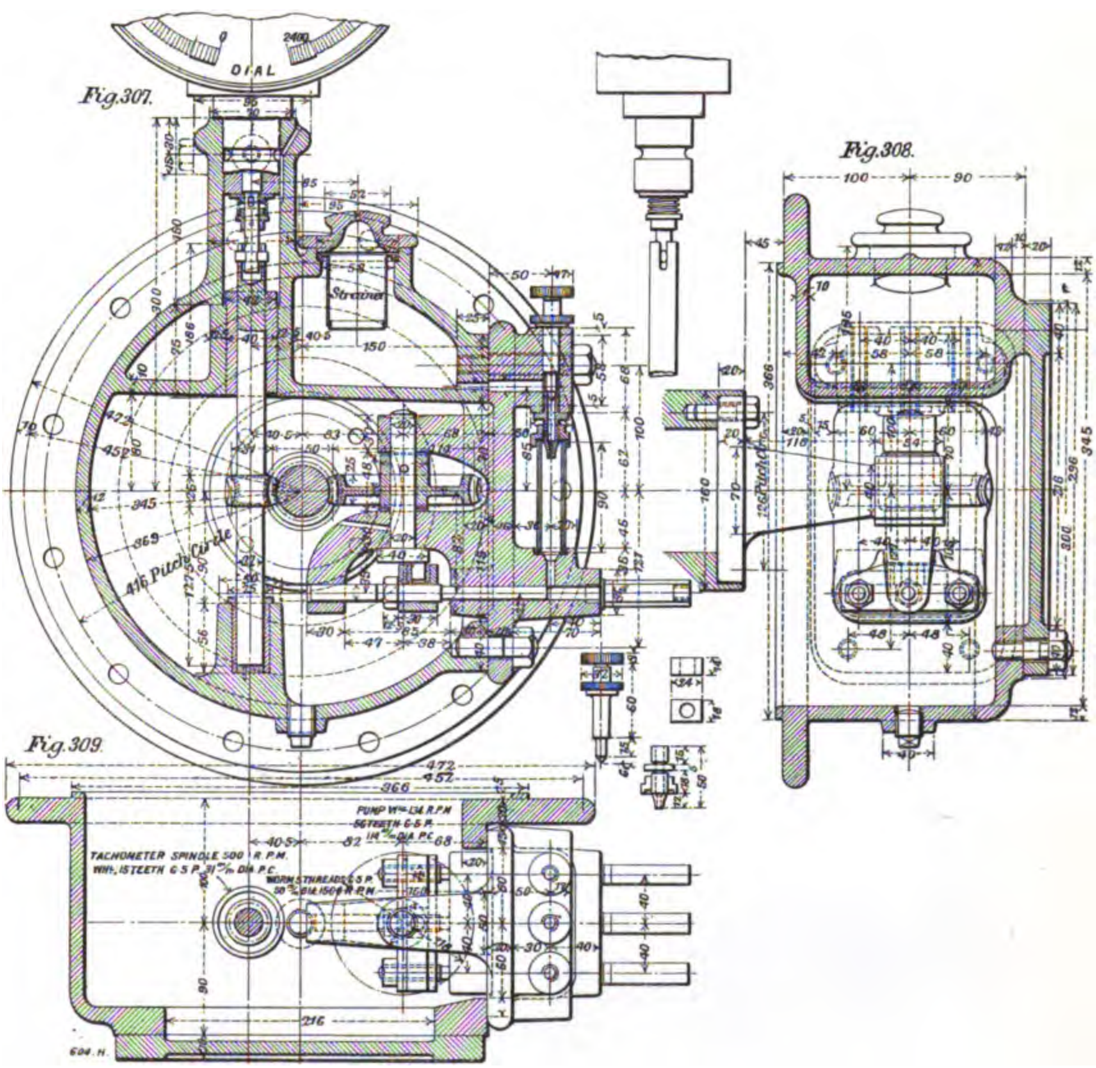
Figs. 300 to 306. Main Bearing for Zoelly Turbine.

metal, and are not made to swivel. In the low-pressure pedestal is held the thrust block, as shown in Fig. 296. This, of course, is split horizontally, but the parts make a metal-to-metal joint, and are registered against relative endwise motion by a small steel key fitted across the joint at each side. Moreover, the complete block has a groove turned all round it, into which fit seatings formed on the top and bottom parts of the pedestal, so that the block is held solid with the latter. The thrust grooves are lined with white metal. To adjust the turbine rotor in the casing, the whole pedestal is set into the required position to give the requisite blade clearance, the shaft being compelled to move with the pedestal by reason of the thrust block. When the right position is determined the pedestal is secured by four steady pins, so that it is fixed once for all.

It will be seen in Fig. 308 that bolted to the end of the turbine shaft is a steel extension piece, on which is cut a worm. This gears with two worm wheels, see Figs. 307 and 309, a small one, running at 500 revolutions, being cut on the tachometer shaft, and a larger one, running at 134 revolutions, driving an oil pump for lubricating the bearings, glands, &c. The tachometer shaft extends upwards and drives the instrument spindle by a sort of loose cotter joint, as shown. The whole of the worm gear, the footstep for the tachometer shaft, &c., are inside a casting, and run submerged in oil. The pump is of the reciprocating type, driven by a die in a slotted crosshead, the die being on the crank pin of the shaft, to which the worm wheel is keyed. There are three rams from the crosshead, working in parallel chambers bored in the solid metal of the casting. A sight-feed device controls the inlet to each pump, and the oil is forced out through a non-return valve. To withdraw the whole of the pump gear it is only necessary to undo four nuts, and the casting containing it can then be removed bodily without interfering with any adjustments. The worm gear, &c., can also be examined through another opening, normally closed by a large circular cover.

The stop valve is illustrated in Fig. 310, page 260. The valve itself is a simple single-beat construction, working horizontally, guided by the bore in which it slides. Steam enters by the bottom branch, passes round the port formed in the casting,

across the valve seat, and away to the governor valve, which is bolted to the right-hand branch. The thrust of the valve spindle when closing the valve is transmitted to the latter through a spring, as shown. The spring is placed between a washer on the spindle



Figs. 307 to 309. Tachometer and Oil Pump Drive

and a plate bolted to the back of the valve. Thus the valve cannot be brought into contact with its seating with the violent blow which might occur were the connection rigid. After contact is made the pressure is increased by the compression of the spring until the end of the spindle comes into action and holds the valve

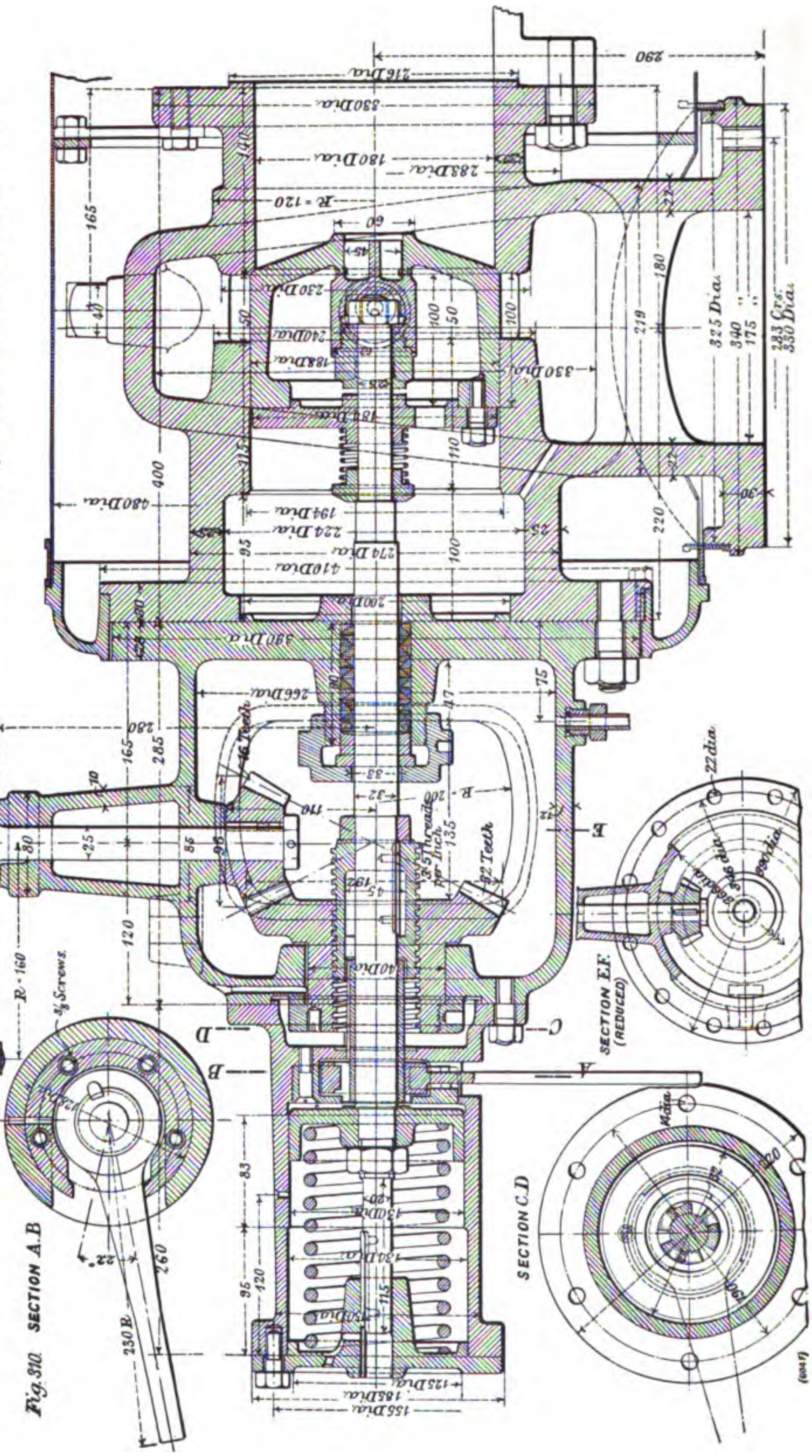
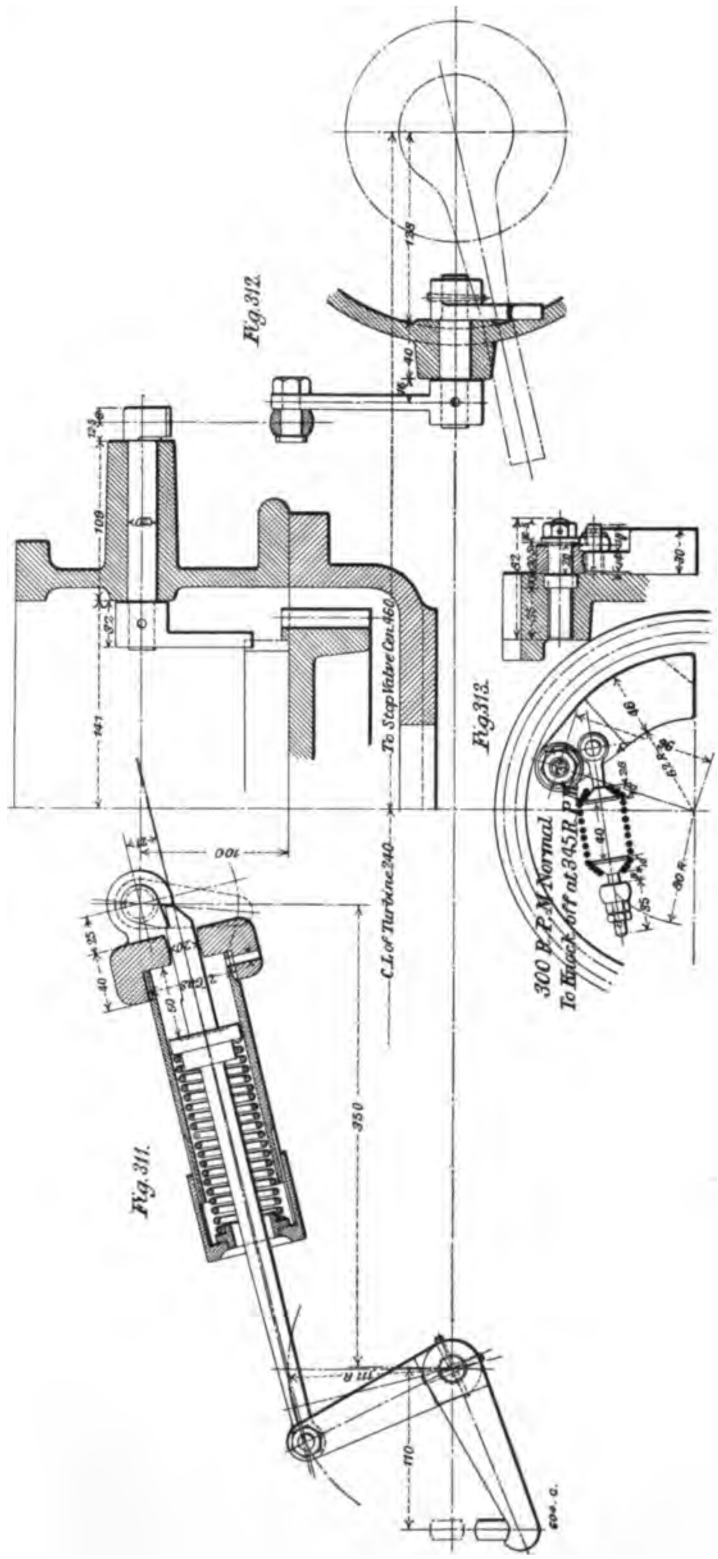


Fig. 310. Stop Valve and Gear for Zoelly Turbine.



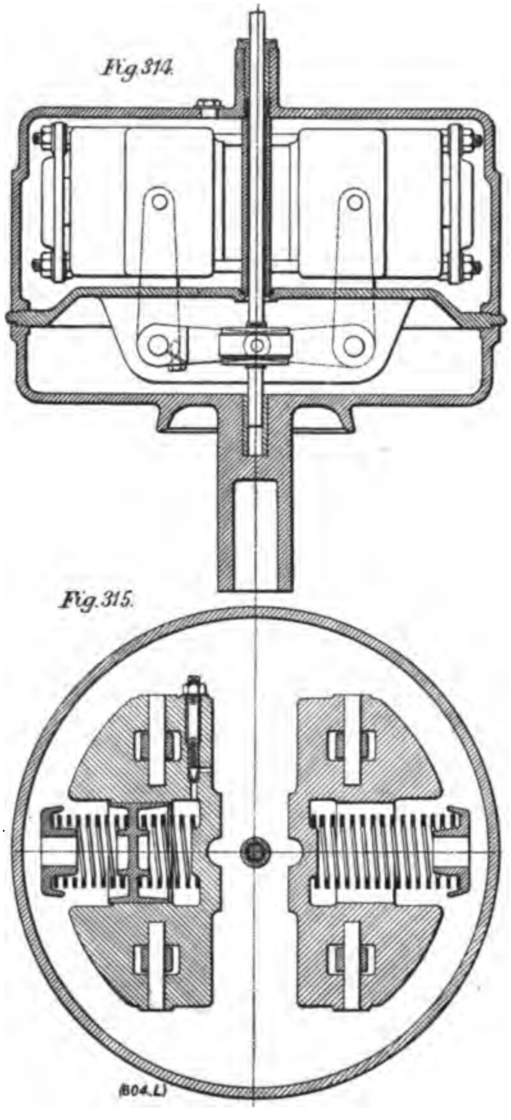
Figs. 311 to 313. Emergency Governor and Gear; Zoelly Turbine.

rigidly to its seat. This provision is all the more necessary as the same valve is used as an emergency valve, and when acting in this capacity, of course, closes at great speed. The arrangement will be understood from the engraving. The normal operation is by means of the large hand wheel. This causes the rotation of a bevel wheel cut with an internal thread, and acting as a fixed nut for a screwed sleeve. The sleeve is prevented from rotation by a spline, and from independent endwise motion by a collar pinned to the spindle at the forward end and a catch sleeve behind. These latter form abutments, through which the sleeve moves the valve spindle.

In a dash pot surrounding the end of the spindle is a piston attached to the latter. Obviously, when the valve is opened, the backward motion of the spindle compresses the spring behind the piston, the thrust passing through the catch sleeve. The spring has a working travel of 2 in., and exerts pressures, at the ends of its travel, of 350 lb. and 535 lb. respectively. The catch sleeve referred to has longitudinal grooves cut along it, and in a certain position these coincide with internal projections in the bore of the screwed sleeve. Normally, however, there is not coincidence, but should the trip lever, shown keyed to the catch sleeve, in Figs. 311, 312, and 313, on page 261, be depressed, the sleeve will be rotated until the two coincide. This is the emergency trip motion, the dash-pot spring then driving the spindle forward independently of the position of the bevel wheel, and closing the valve. The catch sleeve, of course, slides inside the screwed sleeve, the latter remaining as before.

After leaving the stop valve, the steam flows to the throttle valve, which is, of course, controlled by the governor. The governor employed by Messrs. Mather and Platt is that known as the "Chorlton-Whitehead" type. In external form it is a polished cylindrical casing forming a chamber filled with oil, in which the working parts are immersed. The internal mechanism is illustrated in Figs. 314 and 315, which, however, refer to a governor of slightly different type, in which the connection to the valve gear is made from a spindle projecting through the top of the casing. In the case of the turbine governor the connection is made below, for convenience, but otherwise there is no essential difference. The governor is divided horizontally by a centre plate, to a lug on

which the levers are attached. Two levers go upwards to each weight, to which they are pinned, as shown in Fig. 315, which gives a section through the weights. A single lever on each side communicates the motion of the weights to the governor sleeve or spindle. The weights are cored out to receive the springs, the outer ends of the latter being held by washers connected together by bolts passing through the weights themselves. One of the springs is divided into two parts, between which a piston is interposed. This piston fits the bore of the weight, and thus a dash pot is formed, which prevents hunting of the governor. There is an adjustable needle valve controlling a port communicating with the dash pot. The bottom part of the casing, below the diaphragm, is entirely filled with oil, and the top part is about four-fifths full. If the dash-pot valve is wide open, so that the oil can flow freely in and out of the dash pot, the latter is out of action, and the divided spring will behave as one complete spring, the piston merely acting as a washer. The compound spring, under these circumstances, is exactly equivalent to the single spring in the other weight. Both are designed so as exactly to counteract the centrifugal force of the weights in all positions at a constant speed. Thus any variation in speed would move the governor through

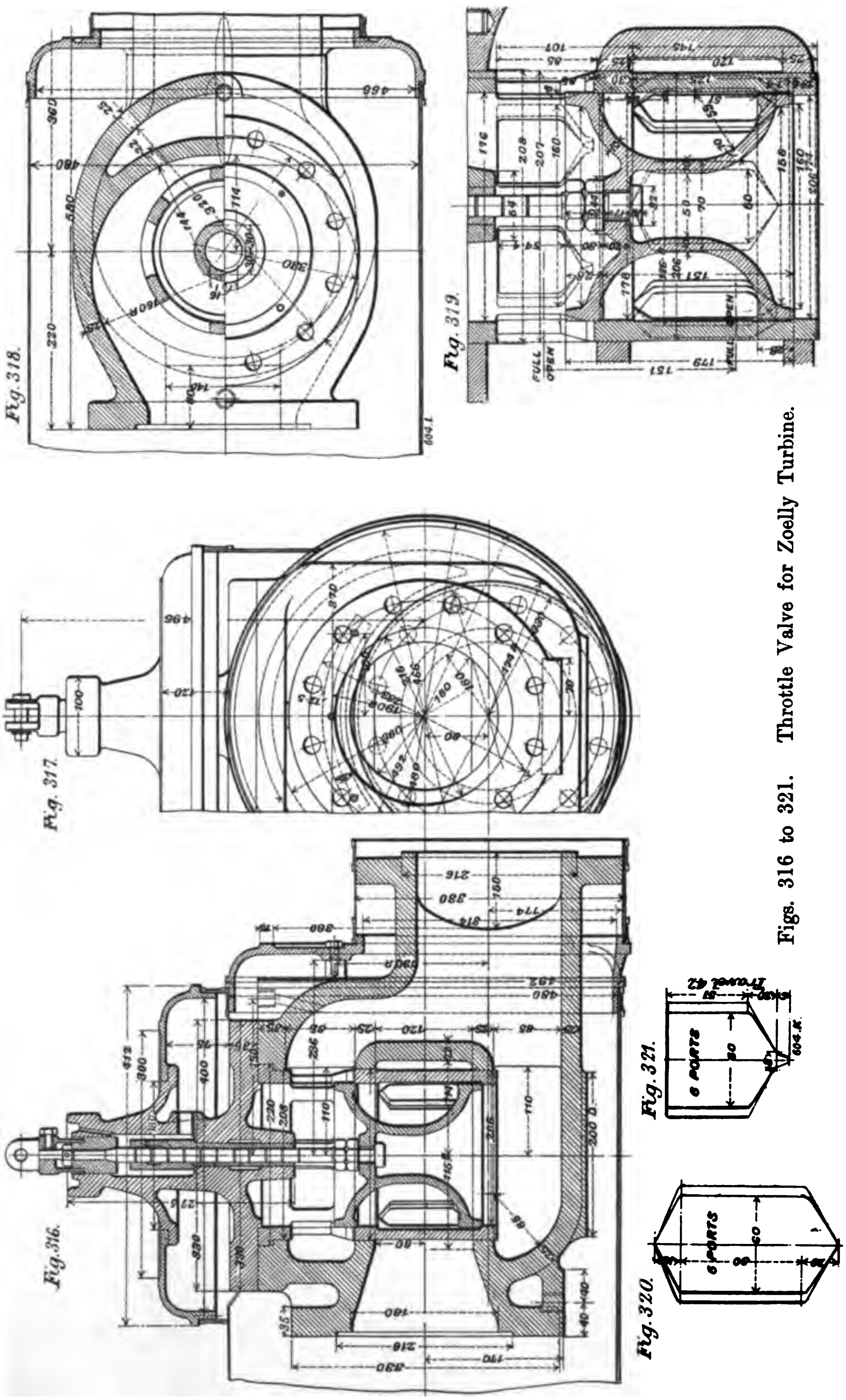


Figs. 314 and 315
Chorlton-Whitehead Governor.

its whole range, and the engine would hunt violently. On the other hand, if the valve were completely closed, the piston would become immovable, thus putting the shorter part of the compound spring out of action. This would virtually stiffen the whole of the governor spring—so much, in fact, that a variation of some 8 per cent. in speed would correspond to the extreme positions of the governor. This is too much for close governing, but a position of the valve can be found in which hunting is prevented and yet the governor will be isochronous, subject to a time lag of two or three seconds. It will thus maintain a nearly constant speed at all loads, which is a feature only possible with an isochronous governor, and yet will not hunt, as such governors always tend to do.

Drawings of the throttle valve are given in Figs. 316 to 321, page 265. Referring to Figs. 316 and 318, the steam enters on the left-hand side, and, passing round the annular passage, enters the valve by half-a-dozen ports, cored in the cast-iron bush which forms the valve seating. The lower and upper ports are shown in Figs. 320 and 321 respectively. The upper part of the valve is a close sliding fit in the bush, while the lower has about half a millimetre clearance all round, and rests on a coned seat when the valve is shut. When it is open, steam passes both upwards and downwards from the central portion; but the main flow is upwards and out through the upper ring of ports. These taper to an elongated point at their lower edge, so that the control may be gradual. When closed the upper ring of the valve laps over the lowermost point of the upper ports by 4 millimetres, and has a further travel of 42 millimetres. It is 176 millimetres in diameter, and balanced as regards steam pressure, while the simplicity of its form precludes troubles from distortion. The spindle has no packing in the ordinary sense of the word. It passes through a well-fitting bronze sleeve of considerable length, and the part within the sleeve is turned with a number of shallow grooves. Any slight leakage which occurs will be in the form of water, and a drain is provided to take it away. The upper part of the spindle carries a small solid piston, which fits the casing well, and serves both as a guide for the spindle and a further check to the escape of steam.

The emergency governor, which actuates the tripping of the stop valve and thus shuts down the machine should the speed



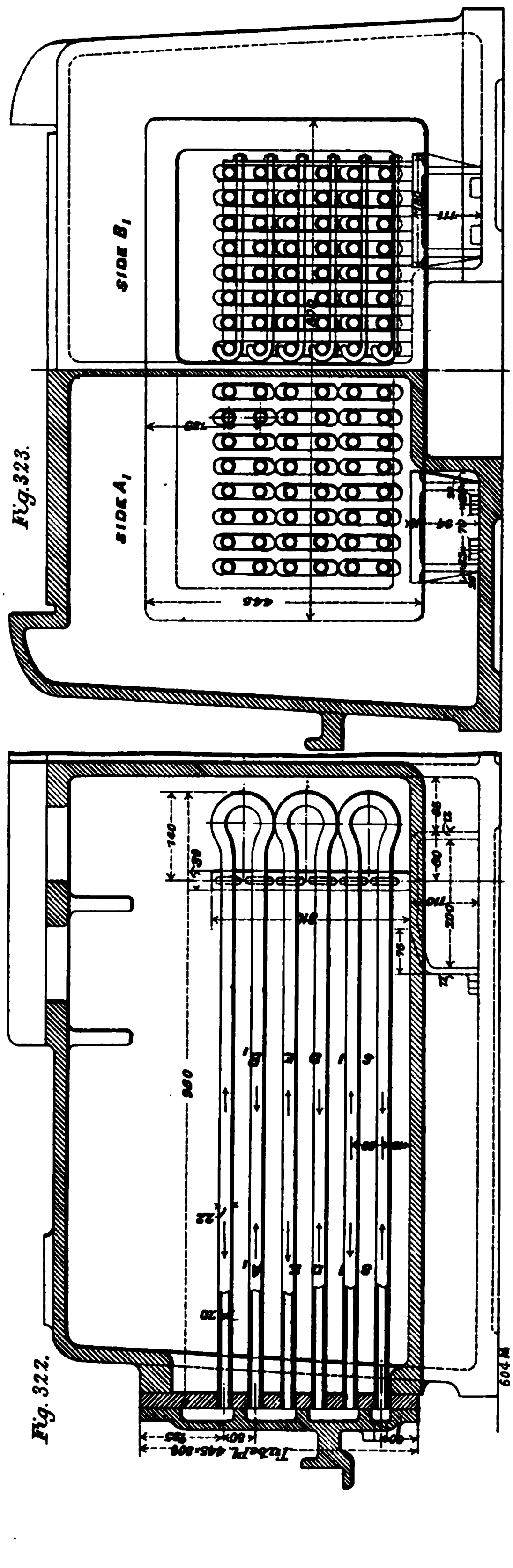
Figs. 316 to 321. Throttle Valve for Zoelly Turbine.

become excessive from any cause, is situated in the base of the casting carrying the Whitehead governor, and is driven by the same worm gear. It consists, see Fig. 313, page 261, of a weight pinned on a plate rotating horizontally, and held in against centrifugal force by a spring. When the speed exceeds a given amount the weight overcomes the spring, and, moving outwards, its hardened end comes in contact with a lever fixed to a spindle extending through the casing. On the outer end of the spindle is a catch, which engages with a notched rod, as shown in Fig. 311. This rod is attached at the other end to another set of levers. When the emergency governor acts, the catch is disengaged and the rod is forced forward by a strong compression spring. In going forward it rotates the levers to which it is attached, and one of these lifts the stop-valve lever and releases that valve in the manner already described above. There is no wear on any part of the emergency governor, as the parts only move in cases of emergency.

The lubrication of the turbine bearings has been referred to in connection with the description of them. The oil, after leaving the bearings, is returned to a large cast-iron tank beneath the high-pressure bearing, where it is cooled. Cooling is effected by means of a system of water pipes inside the tank. The arrangement is shown in Figs. 322 and 323, page 267. Each separate pipe is of U shape, having its ends expanded into the copper tube plate, which forms an end cover for the tank. Outside this is a water box, with diaphragms which compel the water to pass upwards through one vertical row of tubes in series, and downwards through the next. There are altogether forty-eight tubes; they are of rolled brass, 20 millimetres inside diameter and 1 millimetre thick. They are supported at the far end by round rods threaded through bars, the latter serving to space the vertical rows.

Disc Friction.

In impulse turbines the losses by disc friction may be considerable, but really satisfactory and consistent experiments upon this head are still wanting. The loss depends very largely on the amount of clearance allowed on each side of the wheel. Rotating in free air the resistance experienced by a wheel may be two to four times as great as if the wheel is closely encased. In some experiments



Figs. 322 and 323. Oil Cooler for Zoelly Turbine.

a 42-in. wheel, carrying two rows of blades and rotating in free air at 3000 revolutions per minute, absorbed over 70 horse-power.

The large effect of the side clearances on the wheel is due to the fact that a rotating disc acts as a centrifugal pump. It is well known that if an electrically-driven high-lift centrifugal pump is started up against a low head, the fuses on the motor will go, owing to the motor being tremendously overloaded. A turbine wheel rotating in free air works under somewhat analogous conditions, and the power absorbed is not due merely to disc friction, but to the fact that a large amount of work is done in maintaining a radial flow of fluid. Closely encasing the wheel checks this radial flow and reduces the power absorbed.

Dr. Lasche has given a rule for the power absorbed by rotating discs, based on experiments by the Allgemeine Company, which, in English measures, may be written as follows:—

$$\text{Power absorbed in kilowatts} = \frac{A}{5.5} \cdot \left[\frac{\text{R. P. M}}{1000}\right]^3 \frac{d\,l}{V}$$

Where d denotes the diameter of wheel in inches, l the average length of the blades in inches, V the specific volume of the steam and A a constant, which is unity for a wheel carrying one row of moving blades, 1.2 for wheels with two rows, 1.7 with three rows of moving blades, and 2.5 when there are four rows of moving blades. The formula is said to be valid for wheel diameters from 35 in. up to 47 in., and for values of l ranging between $\frac{3}{4}$ in. up to 2 in. The resistance offered by a turbine wheel when at work is, however, quite different from its resistance when motored round. With a wheel at work and with complete admission there is no true windage due to the blades, whilst with partial admission the blades on the active arc are also free from windage. In estimating friction losses allowance should be made for this circumstance.

CHAPTER XXVI.

STEAM TURBINES OF THE PARSONS TYPE.

THE first steam turbine to be developed on a practical scale was that designed by Sir C. A. Parsons, who built his first unit, one of 6 electrical horse-power, early in 1884. The theoretical and mechanical problems to be solved were of extraordinary difficulty, but the inventor recognised from the outset the great possibilities of the reaction type of machine and by an exceptional combination of courage and genius overcame all difficulties. Many of his discoveries are now embodied in machines working on different principles.

A section through a typical turbine of the Parsons type is represented in Fig. 324, Plate XV., whilst Figs. 325 to 328, Plate XVI., represent a general arrangement of the machine. The turbine in question was constructed by the Brush Electrical Engineering Company, Limited, Loughborough, and was designed to develop 1500 kw. at 1500 revolutions per minute, taking steam at 175 lb. (gauge), superheated to 150 deg., whilst the designed vacuum was 27 in. Using the by-pass, the load may be increased to 2000 kw. for an indefinite period, the limit being imposed by the alternator. It will be seen that the rotor is made with four main diameters instead of the three generally employed. This practice allows of a better correlation between the steam-way and the steam volume, and thus gives a more uniform steam velocity, than the other method, without unduly increasing the length of the low-pressure blades. Steam enters the casing at the annular port A, Fig. 324, and flows away to the right, passing through eight "expansions" or groups of blading on its way to the condenser. The condenser is situated immediately beneath the turbine, and is connected to it by means of a copper bellows pipe, shown in Fig. 325, Plate XVI. The end thrust on each of the four drums is balanced by a dummy piston, of the mean diameter of the blading, these dummies being

shown to the left of the port A. Each drum is balanced separately by a pipe, which equalises the pressure acting on the dummy with that acting on the corresponding drum. These equalising pipes are shown in the illustration, and it will be noted that they are all outside the body of the casing, and not cast as ports through it. The latter practice was favourable to distortion of the casing under steam, a point which has to be rigidly guarded against. The equalising pipes, moreover, are either curved or fitted with expansion pieces, so that they may yield freely under the expansion of the casing. Glands are not used for these on account of the chance of their sticking. The by-pass steam for overloads is admitted around the port B, and the first drum is made slightly smaller in diameter beyond this point, in order that end balance may be obtained when the by-pass is in use.

The rotor body is made of a single steel forging, machined inside and out. At the high-pressure end it is forced into a recess turned in a heavy flange forged solid with the end shaft, and the bolts holding these parts together also hold the ring forming the third and fourth dummy pistons. The low-pressure end shaft is also flanged out, registered, and bolted to the rotor, as shown in Fig. 324. In the early days of steam-turbine construction there were not unfrequent cases of trouble that occurred through the end shafts working loose in the rotor. The former practice was to shrink or force them into place, but the high temperature of live, or especially of superheated, steam caused the spigot ends to come loose through differential expansion. In the turbine illustrated it will be noticed that any differential expansion that tends to occur will act to tighten the connection between the shaft and rotor, as at the high-pressure end the heated rotor forms the inner member. At the low-pressure end the likelihood of trouble is not so great, because the temperature differences are less. Moreover, all bolts used are made a driving fit into their holes, and the nuts are locked by riveting over the ends of the bolts. The rotors for the smaller sizes of machines are made in a somewhat different manner, and will be referred to later. Another solution of the same problem will be found in the description of the Willans-Parsons turbine, below.

The upper part of the casing is hinged to the lower at each end, so that the machine may be readily opened and closed

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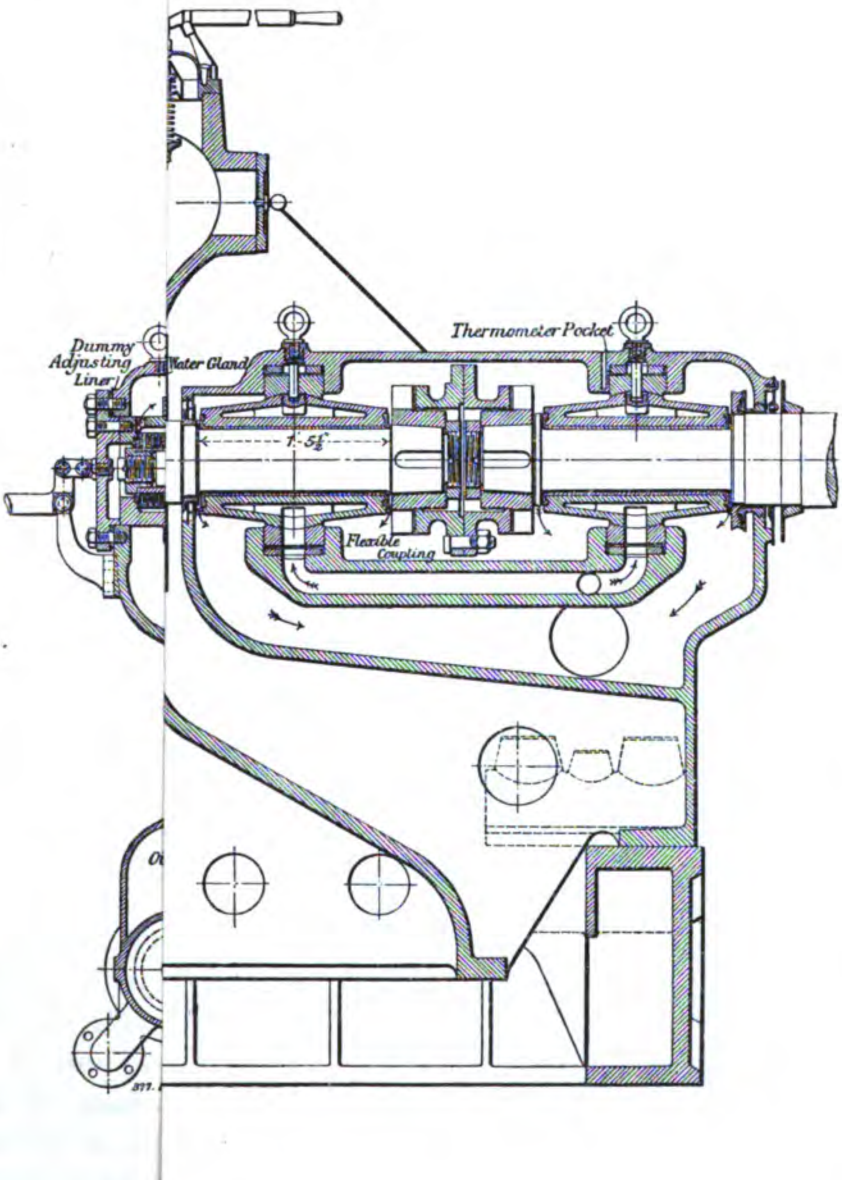
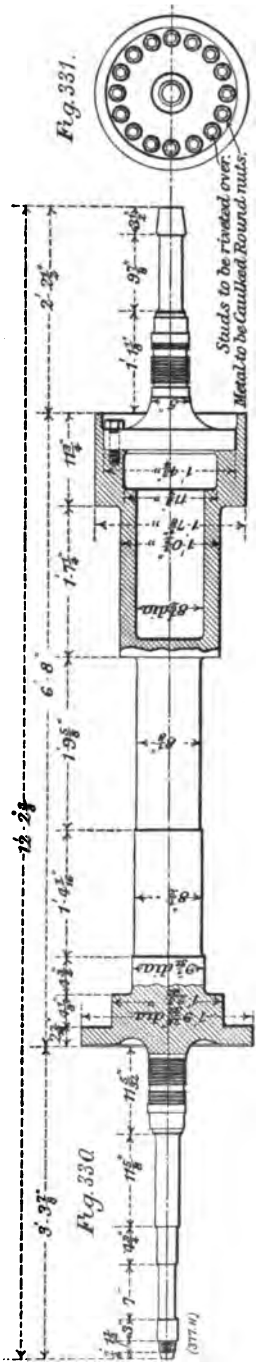

Dummy
Adjusting
Liner
Water Gland
Thermometer Pocket
1'-5½"
Flexible
Coupling

without getting out of register. These hinges can be seen in Figs. 326 and 327, Plate XVI. Each part of the casing for turbines of over 500 kw. is built up of three castings, so as to lend itself to rapid and accurate machining. The castings register together, and are permanently fastened together by means of studs, as shown in Fig. 324. After planing the joint and rough boring, the horizontal joint is scraped and the boring and grooving are completed, special care being taken that the axis of the bore is in the plane of the horizontal joint. The casings are drilled to jig, so that perfect correspondence is obtained between the holes in the adjacent flanges. The exhaust end of the casing is bolted to the turbine bed plate, while the foot at the high-pressure end is free to slide between machined guides, so as to allow the longitudinal expansion and contraction to take place without restraint.

In the end casings are cast inspection holes, normally closed by covers, by removing which it is possible to see the ends of the rotor. The principal use of the orifices, however, is to enable any contact of the blades or dummies to be easily heard when the rotor is slowly moved round. An exhaust-relief valve is fitted, as shown in Fig. 324, which opens automatically should the pressure at the exhaust end of the turbine ever exceed that of the atmosphere. This valve can be operated by hand, by means of a lever, which, when depressed, destroys the vacuum and rapidly brings the turbine to rest when shutting down. A large oil tank, closed by a removable cover, is formed in the turbine bed. From this tank oil is forced by the pump to all bearings of both the turbine and the generator. Details of this pump are given below. The return of the oil from the generator bearings to the tank is by means of a 4-in. wrought-iron pipe cast in the bed plates. At the junction of the bed plates the pipe ends are recessed a little into the faces and expanded into the holes. The joint is made by a ferrule of lead, and proves perfectly oil tight in practice. Thickly shellaced brown paper also makes an effective oil-proof joint.

In this turbine a so-called "water" gland (Fig. 329) is used to pack the shaft where it emerges from the casing. Both ends of the rotor are in communication with the condenser, so that leakage of air inwards is what has to be provided against. It consists of a brass disc forced upon the shaft, the circumference of the disc being

flanged and turned with fine serrations. This rotates in a chamber with about 15 mils clearance all round. There is water in the chamber which is thrown to the circumference by centrifugal force, and effectually seals the clearance space. The smaller flange formed on the outer end of the boss of the disc acts as another seal in series with the first, but its main function is to prevent water escaping outwards from the gland. The circumference of the chamber in which it revolves communicates at the lower side by a groove with the larger chamber, so that water is returned to the latter.



Figs. 330 and 331. Rotor for 500 Kw. Turbine.

Figs. 330 and 331, show the type of rotor used for machines of 500 kilowatts and less. The rotor is a steel forging, as in the larger type, but, as will be seen, the high-pressure end is forged down to form the shaft, so that no possible source of looseness exists at this end. The low-pressure end shaft is flanged out, and fixed to the rotor by studs. The nuts on these studs are locked by burring over the ends of the studs. Moreover, to prevent the studs themselves from unscrewing, the facing for the nut is recessed $\frac{1}{8}$ in. into the metal of the flange, and made just large enough to clear the corners of the nut. When the latter is screwed up, the metal of the flange round the facing is caulked over, so as to catch several corners of the nut.

Figs. 332 and 333, pages 274 and 275, show the design of the stop, throttle, and by-pass valves, and a photograph of the throttle valve and its seat is reproduced in Fig. 334. The stop valve is horizontal and is arranged to act as an emergency valve when tripped by hand or by the emergency governor. By means of a by-pass

Fig. 327.

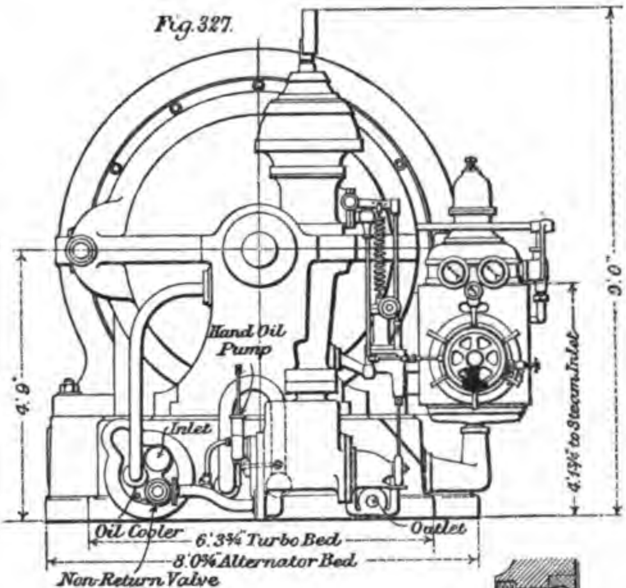
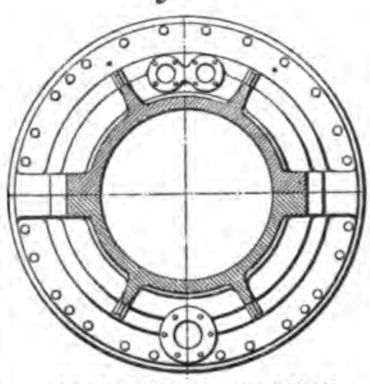
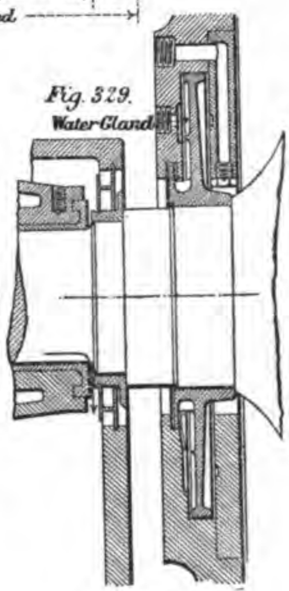


Fig. 328.



CROSS SECTION THROUGH A SMALL CASING

Fig. 329.



To Face Pa

formed on the valve itself the stop valve is practically balanced at the moment of opening. As a stop valve it is actuated in the usual way, by a hand wheel and screwed spindle, by which it can be opened and closed. The nut, however, is not rigidly fixed to the hand wheel, but is formed as a separate sleeve, free to slide axially in the hand wheel, with which, however, it must always turn, whatever its axial position. The sleeve is kept in position by a trigger connected to the emergency-governor gear and to a hand lever. When the trigger is withdrawn, the sleeve is free to slide with the spindle as the valve closes. When used as an emergency valve, the closing force is supplied by a special steam cylinder. Whatever the position of the emergency trigger, the stop valve can always be closed positively by use of the hand wheel. The chief cause of failures of emergency valves to act has been owing to the valve sticking, through lack of use; and the great advantage of combining the emergency valve with the stop valve as far as possible is that, owing to the necessary use of the valve in the latter capacity, it will be found free and in good working condition when required to act as an emergency valve. The steam tightness of the spindles is secured by making them long and grinding them to fit their guides. No soft packings are used.

The throttle valve, illustrated separately in Fig. 334, page 276, has its seatings formed at the lower end of a long cylindrical casting, so that they shall be affected as little as possible by any distortion of the valve chest under heat. Both the throttle and the by-pass valves are entirely automatic, being controlled from the governor through steam relays. Both valves are of cast iron, and are double beat. The relay for the throttle valve, shown to the right in Fig. 332, is of the type originated by Sir C. A. Parsons, in which the governor controls directly a small plunger valve, forming the exhaust valve of the relay cylinder, to which steam is constantly admitted. The plunger valve has also superposed on it an oscillating motion derived in this case from an eccentric on the end of the oil-pump spindle (see Fig. 340, page 280). When the plunger valve closes the relay-cylinder exhaust, the steam pressure rises under the piston fixed to the valve spindle and raises it against the force of a strong spring, which at once closes the valve, when the relay-cylinder

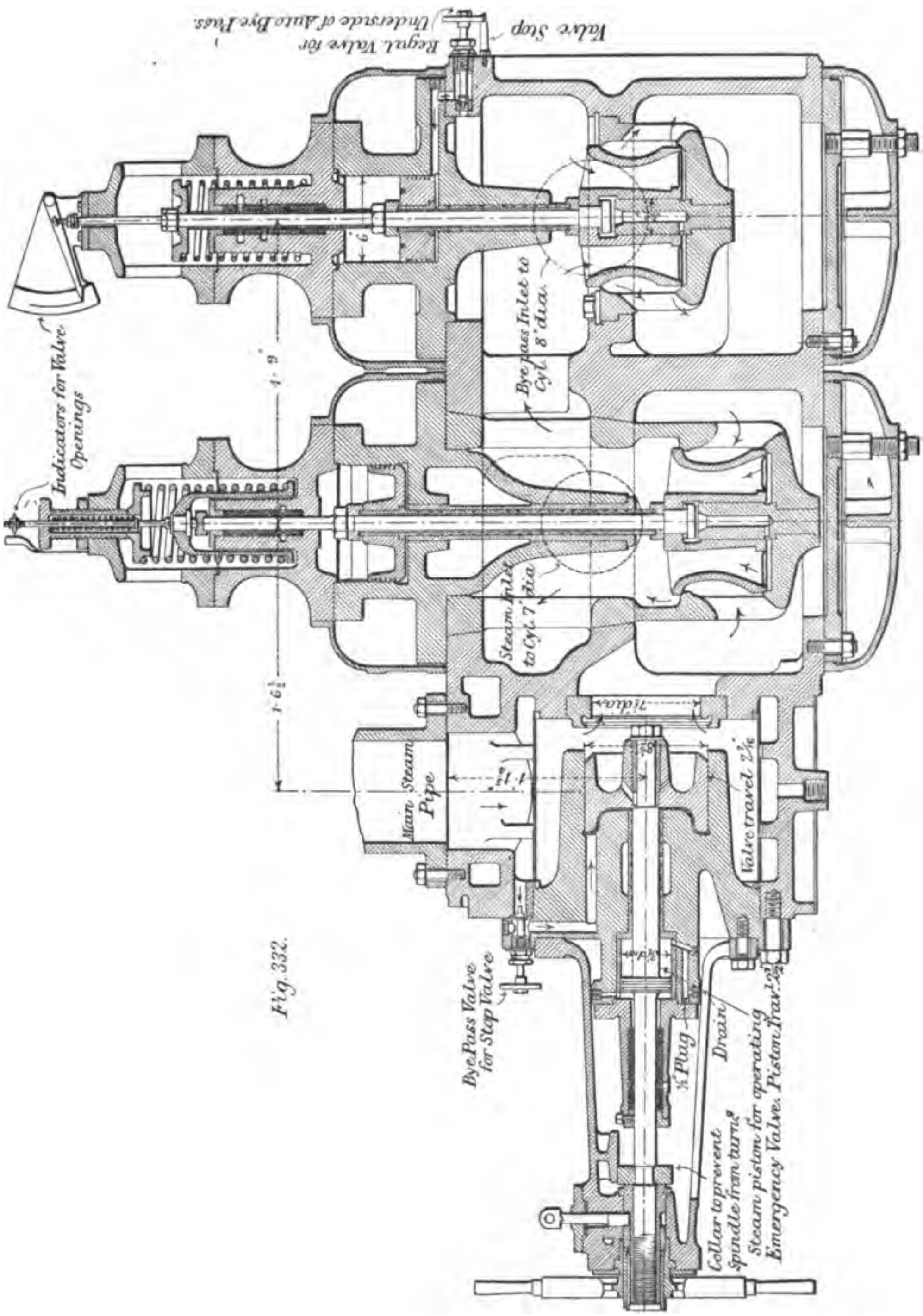


Fig. 332. Stop Valve, Governor Valve, and Overload Valve.

Fig. 333.

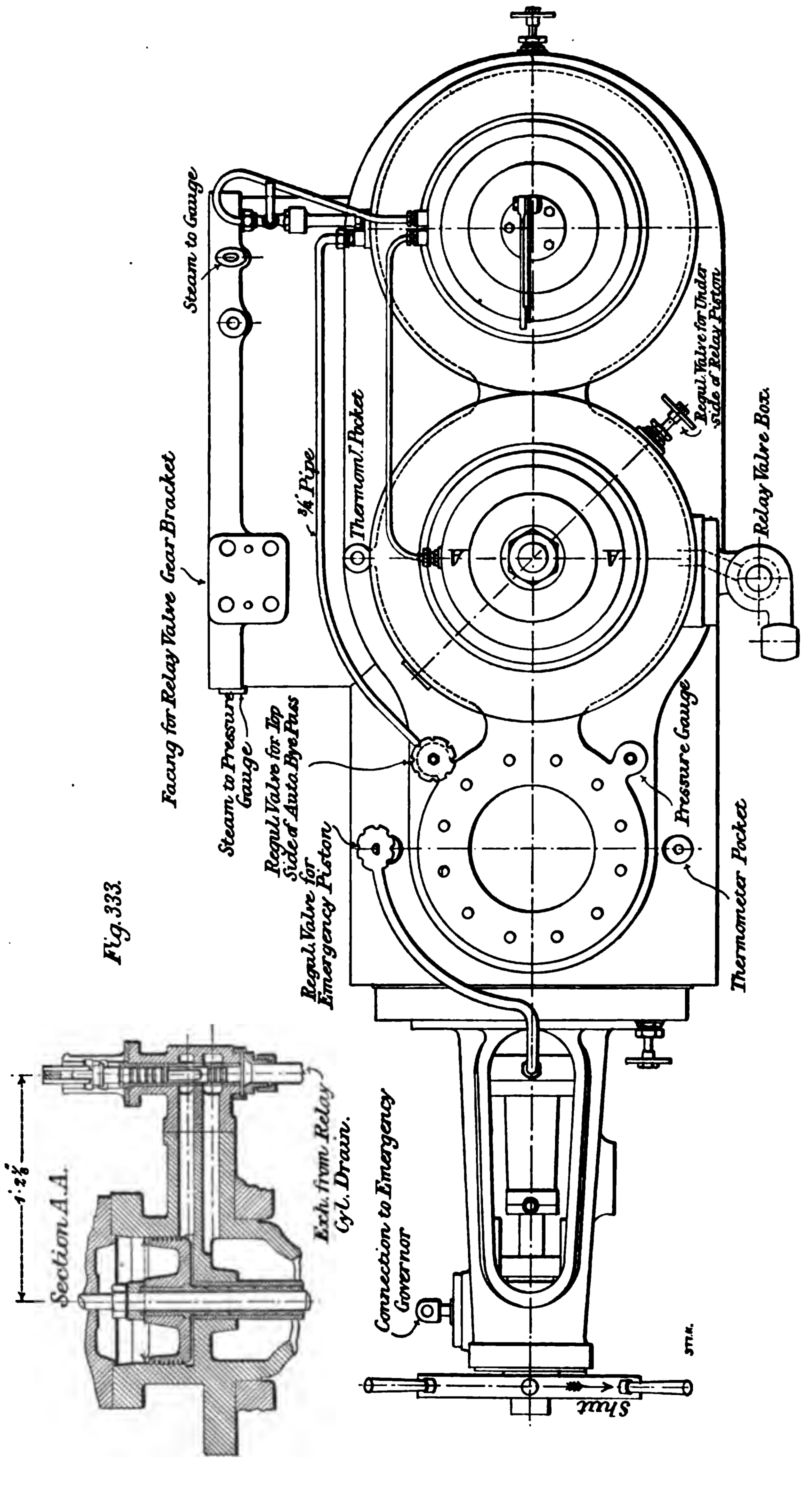


Fig. 333. Valve Box for Brush-Parsons Turbine.

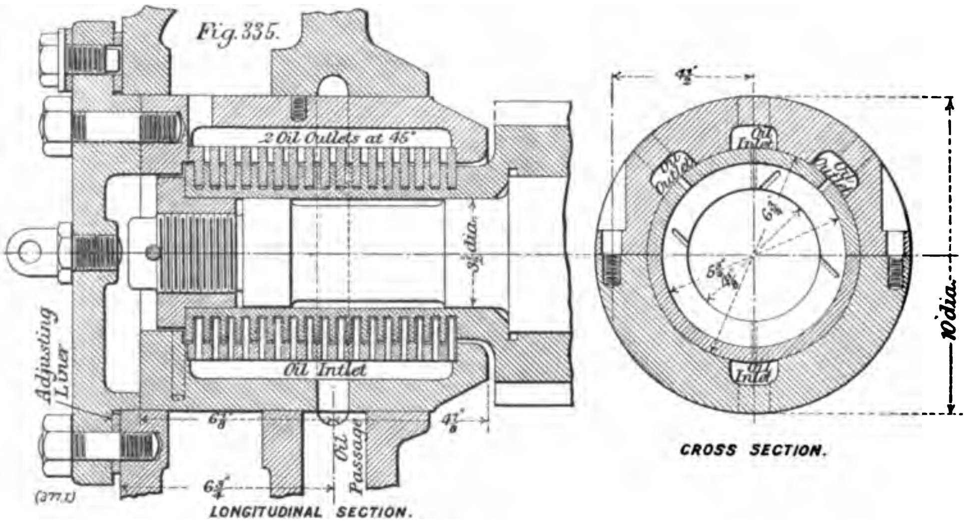
exhaust is again opened through the movement of the plunger valve. Through the periodic motion of the plunger valve, the main valve also acquires a periodic motion, and admits steam to the turbine in puffs, which are of longer or shorter duration according to the load.

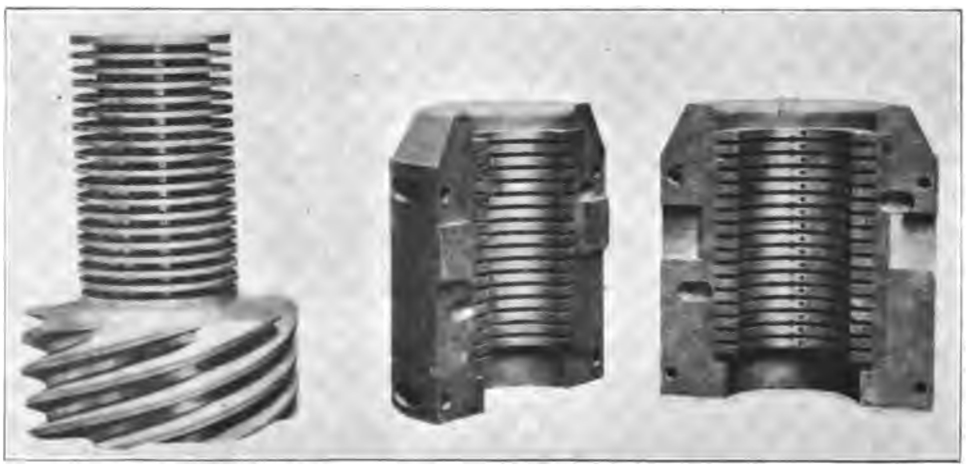
In order to avoid deterioration of the relay spring by subjecting it to high temperatures, it is arranged to be entirely outside of the steam space. The by-pass valve is controlled in a similar manner to the throttle valve, and comes into action automatically on overloads. The weight of the valve box is supported by a strong spring contained in a bracket attached to the turbine underbed, thus relieving the turbine of any strains due to its weight.

Fig. 334. Governor Valve; Brush-Parsons Turbine.

The simple and very efficient form of thrust block, for taking up what slight end thrust there may be on the spindle, and for accurately locating the position of the rotor longitudinally, is shown in Figs. 335 and 336. This form was adopted after numerous trials of various types of thrust bearing at high speeds. Since it is essential that the rotor should have an extremely limited amount of end play, thrust blocks for reaction turbines are generally made in halves, joined along a horizontal plane, the upper half being free to slide relatively to the lower one. The lower half is then arranged to take any end thrust on the rotor directed towards the thrust bearing, and the upper half that directed from the thrust bearing. There are thus, in effect, two independent thrust blocks with two sets of adjusting gear. In

the case illustrated, however, the upper and lower halves of the thrust block are permanently fixed together, thus doubling the effective bearing surface and abolishing one adjustment. The running collars are of steel; the stationary collars are of bronze turned so as to allow the running collars 0.002 in. to 0.003 in.





Figs. 335 and 336. Details of Thrust Block; Brush-Parsons Turbine.

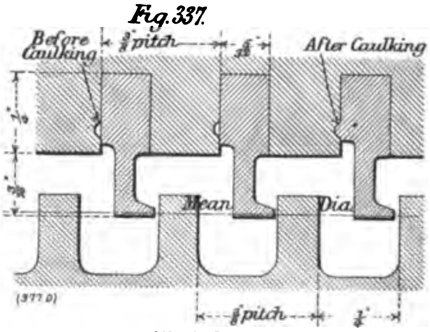
play. At two points on the inner periphery of each stationary ring, oil is delivered under a small pressure, and, aided by the centrifugal action of the rotating ring, makes its way to the outer periphery of the bearing, and is drained away at two points on the upper half of each ring. The delivery and drain holes and passages will be easily followed in Figs. 335 and 336, above.

The chief function of the thrust block is to fix the longitudinal position of the rotor in the cylinder, and thus the "dummy clearance." The method of adjusting this is simple and positive. The thrust block, being securely bolted to its flanged end cover, is moved by means of the lever and link, shown in Fig. 324, towards the left until the dummies come into contact. In that position the distance between the thrust-block cover and end of the cylinder face is then accurately gauged, and an annular liner of that thickness, less the desired dummy clearance, inserted. On tightly screwing up the thrust-block cover the rotor will be set in the required position.

The inverse problem of measuring the dummy clearance at any time is equally simple; it is only necessary to slack back the nuts on the thrust-block cover, and pull back the thrust block till the dummies come into contact. The thickness of feeler which can be inserted between the cover and liner is an accurate measure of the dummy clearance. This operation can be done so quickly that it is easy to measure the minimum dummy clearance under practically any working condition.

The dummy pistons have each a diameter equal to the mean diameter of the blading on the corresponding section of the turbine. They are packed against steam leakage by so-called labyrinth packings, one of the most ingenious of the many ingenious devices originated by Sir C. A. Parsons. The theory of this packing is dealt with elsewhere, but its characteristic features are well shown in Fig. 337. The collars, seen in the lower part of the illustration, are turned on the edge of the dummy piston, and run just out of contact with the L-shaped pieces caulked into the casing. These pieces are usually of drawn brass, but sometimes for the high-pressure dummy, where superheated steam may be met with, they are of drawn nickel steel. They are inserted in lengths of 2 in. to 4 in., and held by caulking, as shown. The shorter the length in which these are inserted the less they are likely to be

Fig. 337.

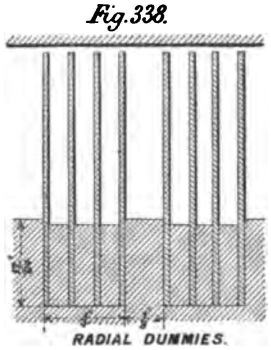


AXIAL DUMMIES.

affected by differential expansion. It will be seen that the steam to leak over the dummy piston has to pass through each of the narrow clearance spaces at high velocity, and as it loses all this velocity immediately, and has to acquire more to pass the next collar, its pressure is rapidly reduced, and the quantity which ultimately escapes is small. Another type of dummy packing adopted for the low-pressure dummy where the large specific volume of the steam makes leakage relatively less important is shown in Fig. 338. In this, although the contractions of flow are more numerous, there is more or less a straight run for the steam.

For the reception of the blades parallel grooves are bored in the casing and turned in the rotor, and the side walls of the rotor grooves are corrugated, as shown in Fig. 339. In the

Fig. 338.



CYLINDER GROOVES.

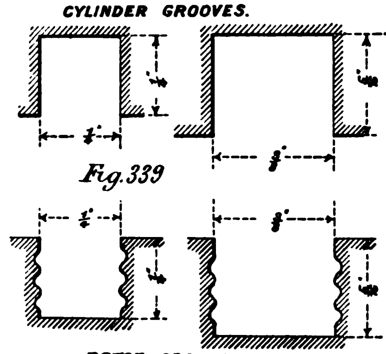


Fig. 339

ROTOR GROOVES.

grooves the blades are inserted alternately with distance pieces, generally known as caulking pieces, and made to the exact shape of the space between the blades and the groove walls: first a blade, then a caulking piece, and so on until the groove is filled with a complete ring of blades. The pieces are then caulked vertically, and their metal spread so as to fill the corrugations in the blade roots and in the groove walls. A lacing strip let into notches in the blades is secured in position by silver solder, and cut at five or six points round the circumference to give flexibility for expansion or contraction under the varying temperatures met with in practice. Such blading is mechanically strong, and not subject to vibration troubles. Each blade is brought to a thinned edge at its tip. Hence should anything occur to bring the periphery of the blade ring into rubbing contact, the thinned tips are quickly worn away, thus auto-

matically relieving themselves. The blades are drawn to the required section and received in the blading department in about 6-ft lengths, which, after having been carefully gauged and examined, are, if found satisfactory, passed into the stores. The individual blades are cut off in a stamping press, which at the same time imprints the two corru-

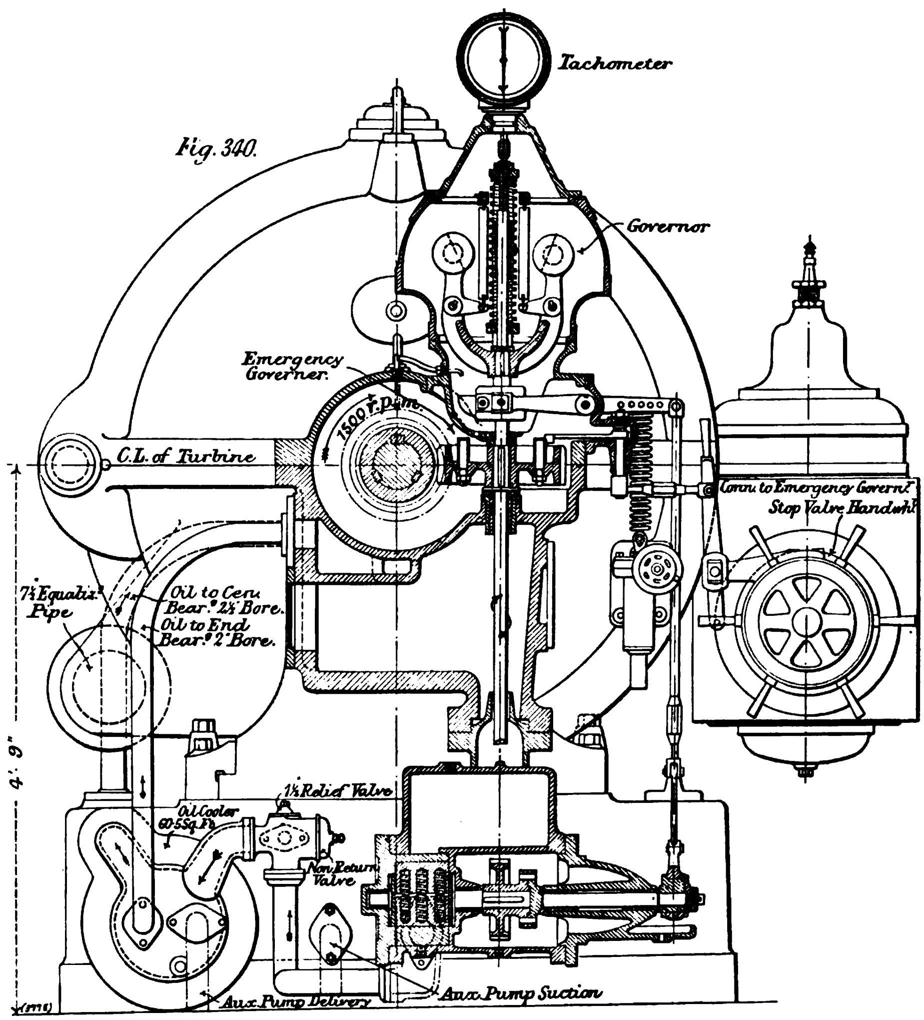
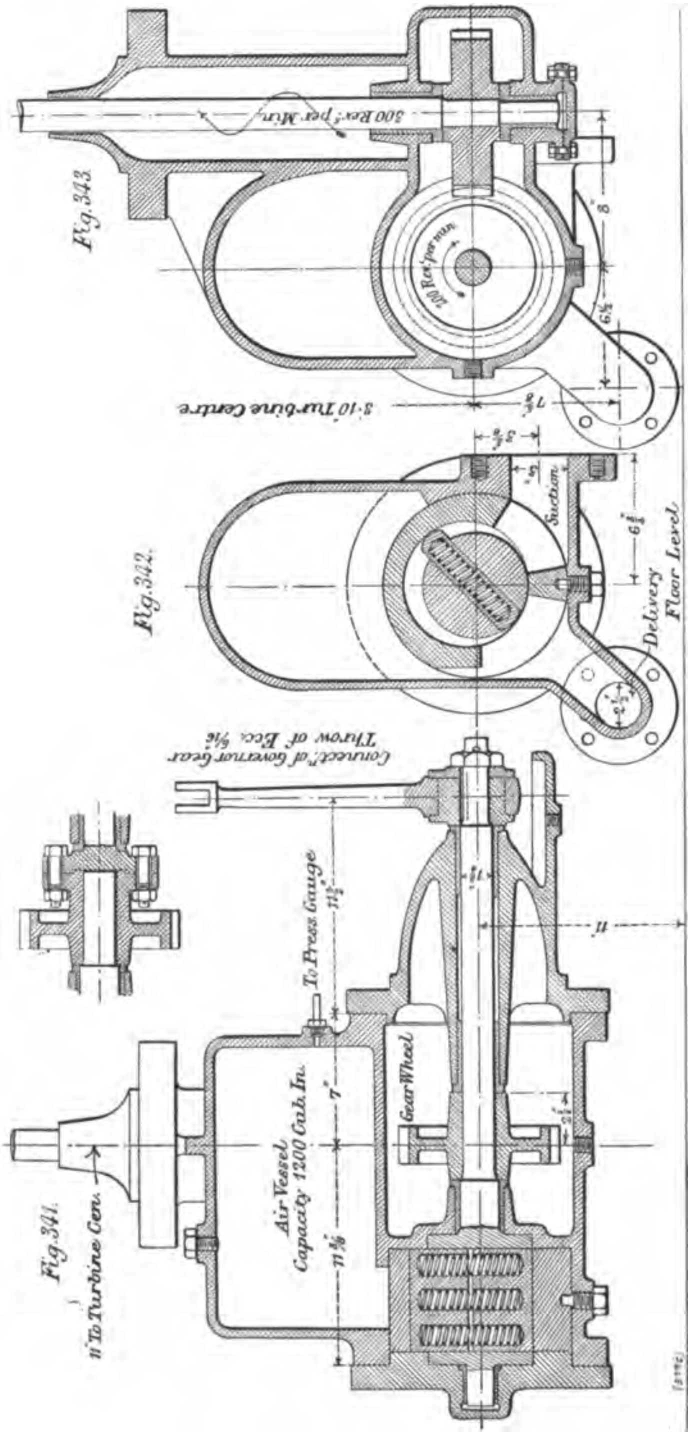


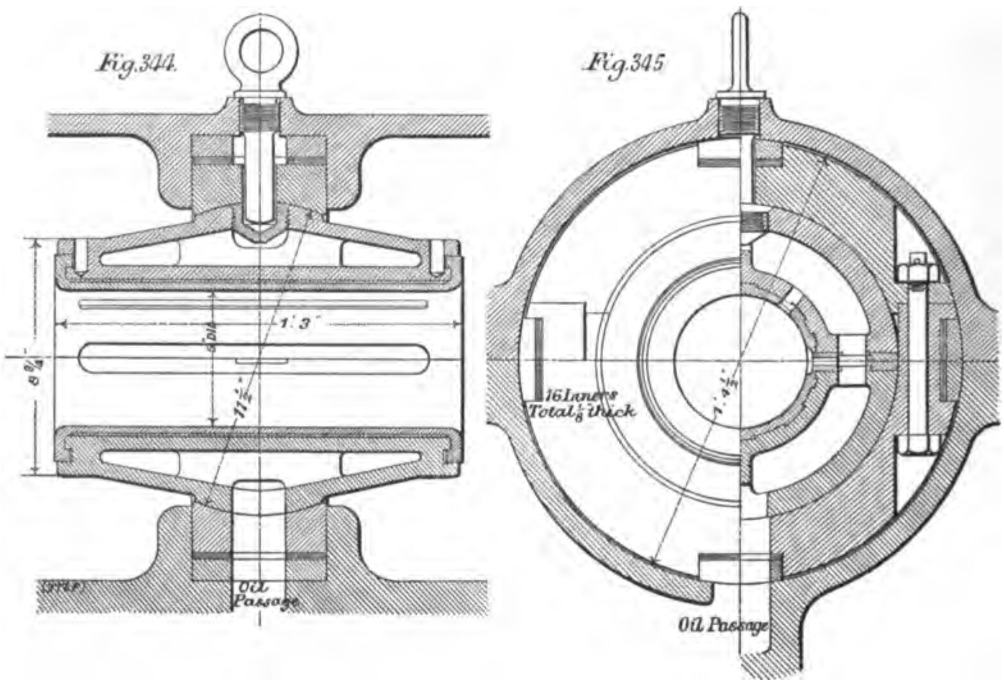
Fig. 340. Governor and Pump Drive; Brush-Parsons Turbine.

gations on the root of the blade. These corrugations are necessary in the case of the rotor blades only, since the casing blades are not subjected to centrifugal force. Experiment shows that the holding force in the blades is not to any great extent frictional, uncorrugated blades being easily pulled out. With corrugations of from 0.02 in. to 0.04 in. deep, however, the blade will pull in two before it draws.



Figs. 341 to 343. Oil Pump; Brush-Parsons Turbine.

As shown in Fig. 340 the governor and the independent emergency governor are carried on a vertical spindle driven from the main turbine shaft by helical gearing. The regulating governor is spring-controlled, and is fitted with knife-edge joints, and the main spring is in halves, coiled respectively right and left hand. The emergency governor, recessed into the helical wheel, is of the knock-out type, and as each weight has its own spring independent of the other weight, the governor is really in duplicate. The



Figs. 344 and 345. Main Bearing; Brush-Parsons Turbine.

emergency governor acts by releasing a trigger, and thus permitting the main stop valve to close automatically.

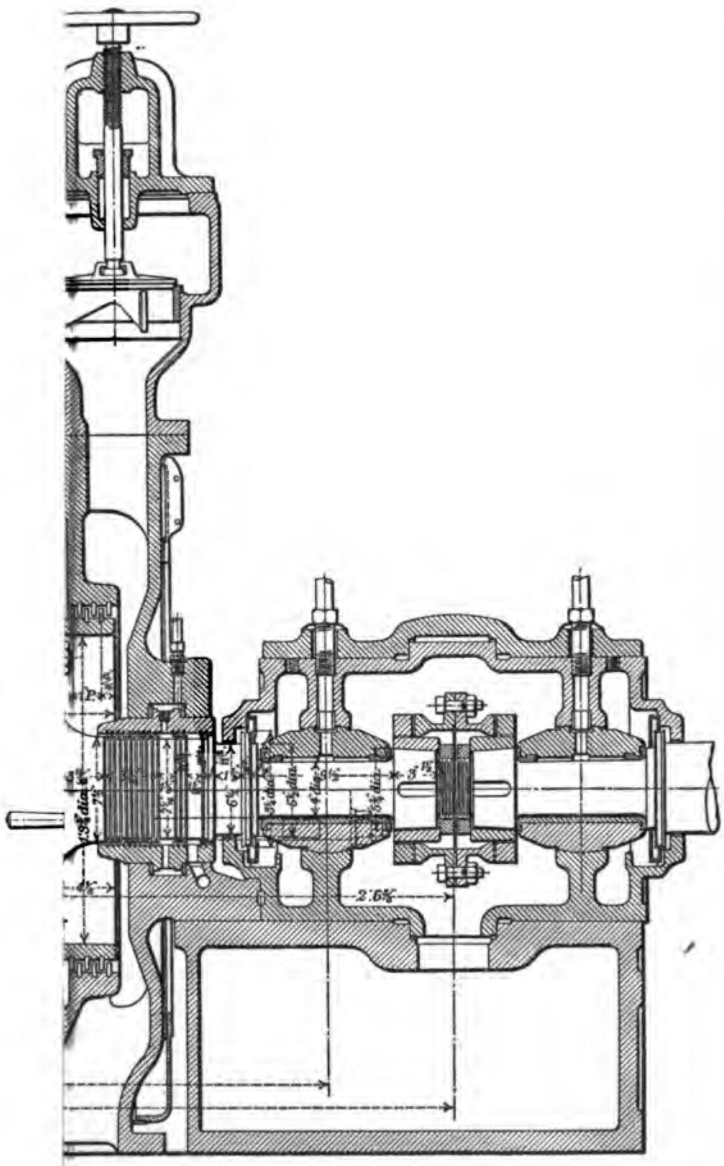
At the lower end of the governor spindle (see Fig. 340, page 280) is a helical wheel, which drives a corresponding wheel attached to the oil-pump spindle. The oil pump is a valveless rotary pump, the arrangement of which is clearly shown in Figs. 341 to 343. The pump draws oil through a strainer, made in duplicate, which may be removed while the plant is running. Oil discharged from the pump passes through an oil cooler to the bearings, and drains thence back to the oil tank.

The standard design of bearing is illustrated in Figs. 344

Fig. 346, PLATE XVII.

TURBINE.

MITED.



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and 345, page 282. The bearing proper, consisting of a cast-iron shell lined with white metal, rests on a spherical seat formed on an outer ring of cast iron, which in turn is supported by the cylinder and keep in a suitably-formed groove. Around the periphery of this ring there is clearance, except where four steel pads, at opposite ends of the vertical and horizontal diameters of the ring, bear against the cylinder and keep. A number of liners of known thickness are included between each steel pad and the ring, and these provide means for adjusting the setting of the bearing in any direction. The steel pads are screwed to the ring, holding the liners in place, and the whole is turned to fit the bore of the cylinder seating.

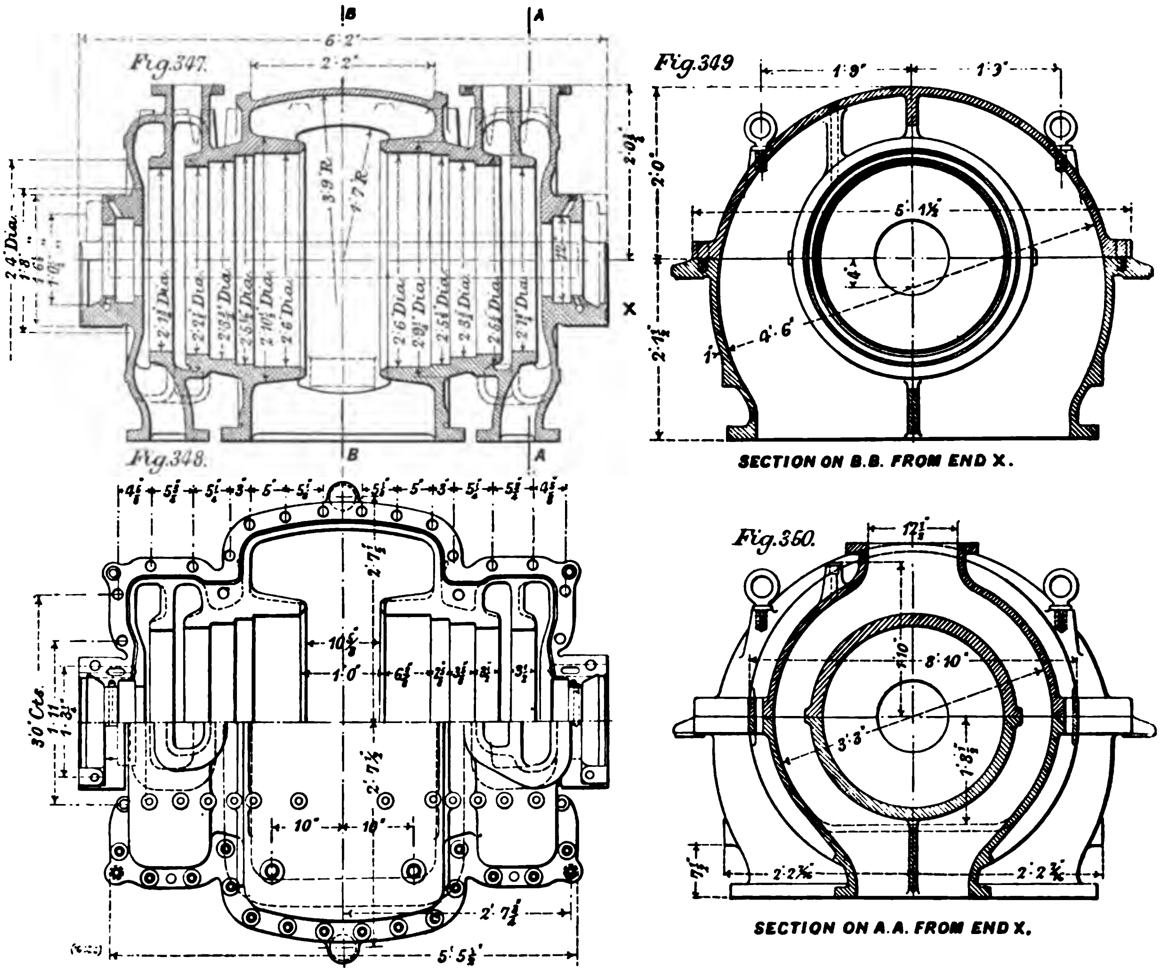
The total thickness of liners on each diameter is equally divided between the ends, and by moving one or more to the opposite end of the same diameter, the bearing centre is moved along that diameter a distance equal to the thickness of the liners moved.

The lubricating oil enters from below, and, passing into the hollow shell of the bearing, keeps it cool, first by conduction, then passing along the oil grooves shown, the oil carries off still more heat, and ultimately escapes at either end of the bearing. The bearing, being made in halves, can, of course, be removed without lifting the rotor.

A 600-kw. exhaust-steam turbine, constructed by the same firm, is illustrated in Fig. 346, Plate XVII., and embodies some important improvements in details of construction.

The system of supporting the turbine casing in its bed plate is worthy of special study. It will be seen from Figs. 346, 347, and 348 that it merely rests on the bed plate, being neither cast with nor bolted to the latter. The bed plate is, in short, an absolutely distinct casting, which has half cylindrical seats bored in it at each end, which in the first place receive the feet or projections by which the weight of the casing is transmitted to the bed. These projections are best seen in Fig. 347, which is a longitudinal section through the casing. The boring of the turbine cylinder and the machining of these cylindrical feet is effected at one setting, so that the finished surfaces are truly co-axial. The bored seats in the bed plate also receive cylindrical shells, in which are arranged the main bearings of the turbine. One of these shells only is clamped in its seat, the other

being free to slide axially, so that both rotor and casing of the turbine are under no constraint so far as longitudinal expansion is concerned, and the latter is, moreover, free to rotate in its seats, and cannot therefore be distorted in the process of coupling it up to the steam pipes and to the condenser. This arrangement seems



Figs. 347 to 350. Casing for Brush-Parsons Exhaust-Steam Turbine.

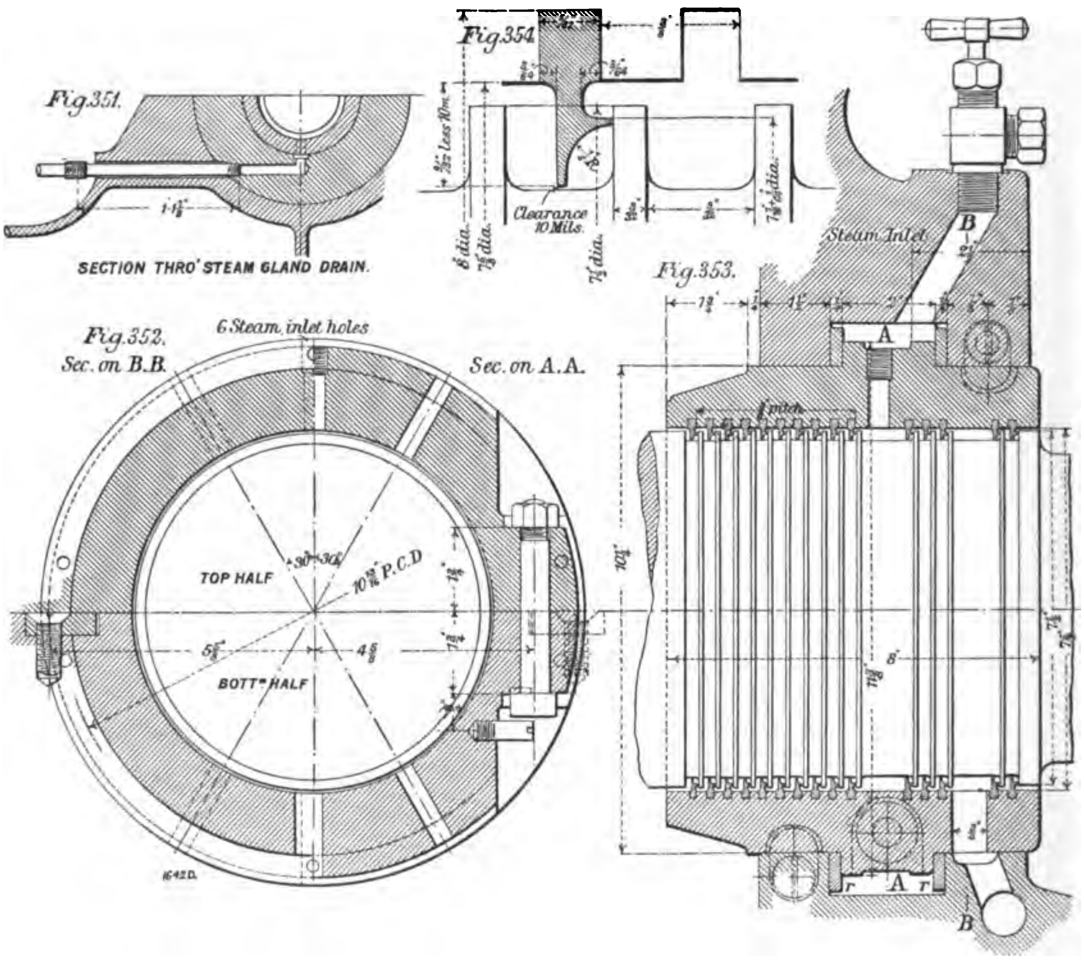
to have many good features, since, in addition to ensuring the absence of strains due to thermal distortion, it gives no loophole for an injudicious or inexperienced erector to pull the casing out of shape when finally assembling the machine. In fact, all the vital parts go automatically into alignment. The weight of the casing is sufficient to hold it securely in place, its rotation being prevented by a stop bearing, on the bed plate at one side. Further

experience with this system of construction has only served to emphasize its good points. On transferring the turned rotor to the casing, after placing the latter in position, careful checking proves that the actual tip clearances nowhere differ from those intended by more than 2 or 3 mils.

A plan of the casing is given in Fig. 384, page 284. The upper half shows the joint with a groove extending completely around it, inside the line of the bolts. A supply of steam at a little above atmospheric pressure is led to this groove, so that should any leakage take place, it will be from the interior outwards, and not a leakage of air into the turbine, which would not only spoil the vacuum with which the economical use of the steam is so closely associated, but the location of the leak would also be very difficult to detect. Fig. 349 shows a section through the exhaust port of the turbine, which measures 3 ft. 8 in. by 2 ft. 2 in. A section through one of the steam ports is given in Fig. 350. The steam-valve box is bolted to the bed plate of the turbine, and connected to the casing by flexible couplings only, thus avoiding the possibility of any distorting stresses being communicated to the cylinder by these connections. A detail showing the construction of one of these couplings is given in Fig. 372, page 291. This is built up out of copper sheet and tubing with flanges brazed on. The connections between the tubing and the expansion plates are both riveted and brazed, the rivets being of copper. These expansion plates are cut from No. 16-gauge sheets, and are also both riveted and brazed to a ring at their outer edges. As will be seen from Fig. 346, a by-pass valve is provided at the coupling end of the turbine, by opening which the first groups of blades at each end can be short-circuited, thus allowing the turbine to take a substantial overload.

The drum is a hollow steel forging having an elastic limit of about 20 tons per sq. in., and an ultimate strength of 36 tons. Each end shaft has a flange 1 ft. $9\frac{1}{2}$ in. in diameter and 3 in. thick, by which it is secured to the drum by fifteen studs screwed to 1-in. standard gas threads. There are also at each flange three symmetrically disposed driving pins, $1\frac{1}{4}$ in. in diameter at the smallest end and turned to a taper of 1 in 48. These are driven into reamed taper holes, and prevent the possibility of any relative motion of the drum and the end shafts. These pins, when home,

are sunk below the surface of the flange, the metal of which is burred over so as to lock them in place. The studs which secure these end shafts are riveted over the nuts when the latter are screwed home, and the flange metal is, moreover, caulked round each nut, thus effectually preventing any possibility of slacking back.



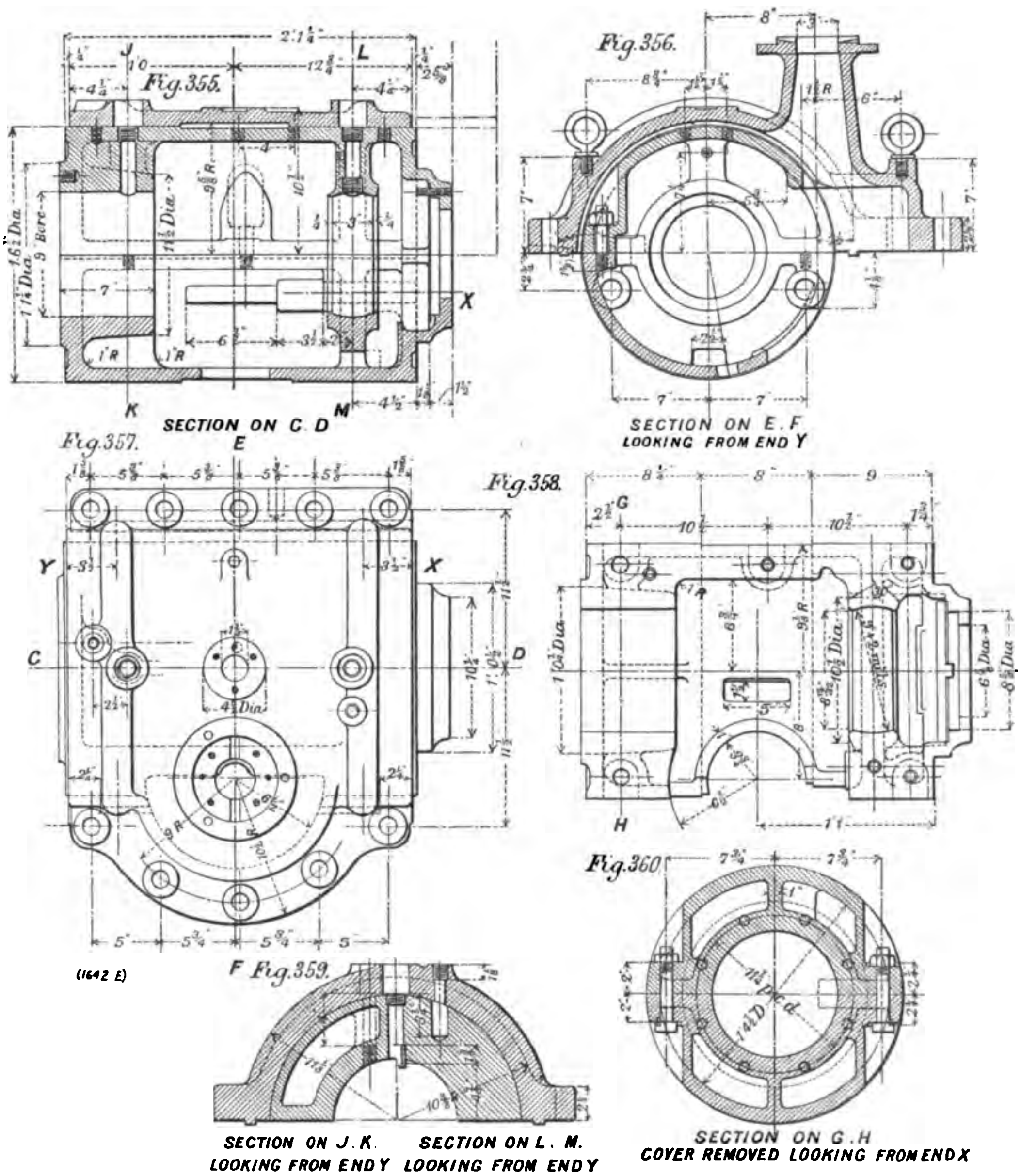
Figs. 351 to 354. Gland for Brush-Parsons Turbine.

The fin shown near the centre of the drum serves to direct the steam flowing from the last row of blades into the exhaust port.

The rotor carries at each end fifteen rows of blades, the shortest being ⅞ in., and the longest 3 in., in height.

The construction of the glands is clearly shown in Figs. 351 to 354, above. The special feature of this gland, as compared with previous patterns, is shown to an enlarged scale in Fig. 354, where it will be seen that the steam is wire-drawn at two points in each

groove in place of at one only. The packing strip, cross-hatched in Fig. 354, nearly touches the groove in the rotor, both at the side and at the bottom. The clearance between the packing strip and

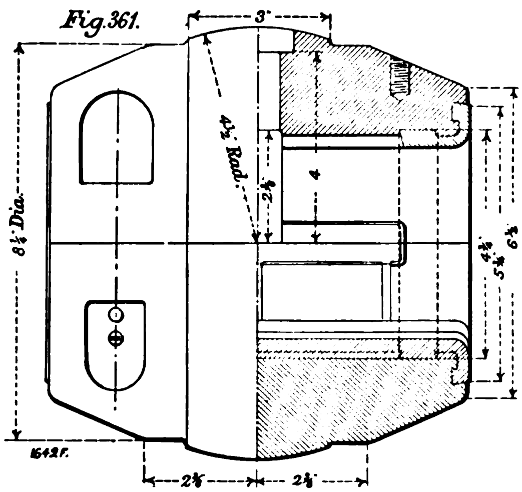


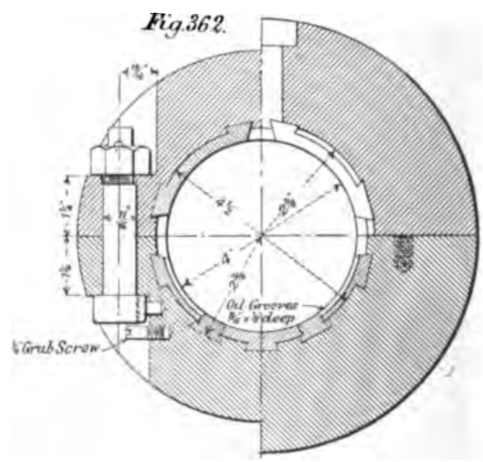
Figs. 355 to 360. Main Bearing Sleeve: Brush-Parsons Exhaust Turbine.

the side of the groove is adjustable. In the case of the gland at the coupling end of the turbine, this adjustment is effected by the micrometer attachment to the thrust block, by means of which the

whole rotor can be traversed axially in or out. At the other gland the clearance is adjusted by shifting the gland bush, the latter being located by the rings shown at *r* in Fig. 353, which are turned up to their final thickness after the erection of the turbine. The gland bushes, it will be seen, are made in halves, bolted together, as shown to the right of Fig. 352. They are prevented from moving on their seats by locking plates secured by countersunk set screws, as indicated to the left of the same figure. A supply of steam is admitted to the gland at A, and the leak off passes through B to the groove cut in the casing joint, as previously explained.

As already mentioned, the spherical seating bearings on which



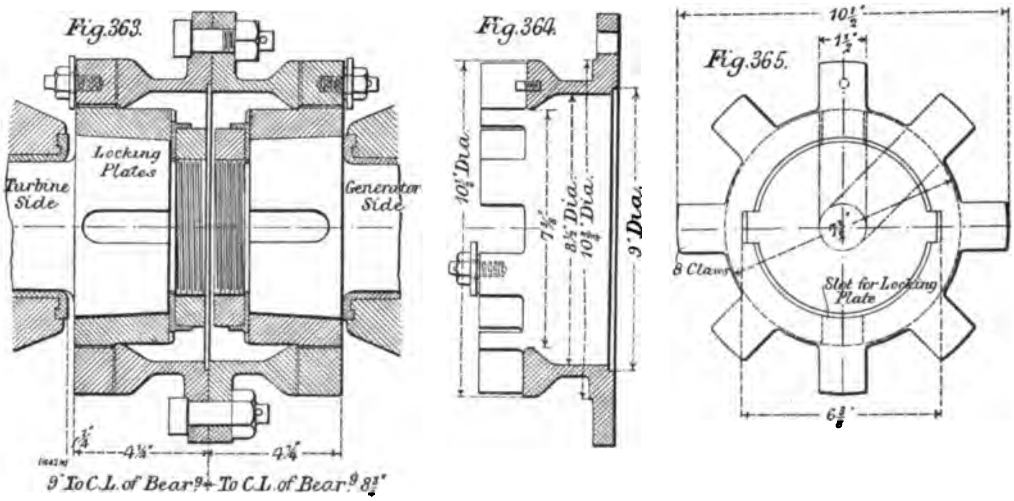


Figs. 361 and 362. Bearings for Brush-Parsons Exhaust Turbine.

the rotor rests are mounted in cast-iron shells, which are turned to a sliding fit for the seats bored in the bed plate of the machine. The construction of the shell for the thrust end of the turbine is shown in detail in Figs. 355 to 360, page 287, though the arrangement of the bearing and thrust block within it is better indicated in Fig. 346, Plate XVII. At one of its ends the shell supports the bearing, and at the other the thrust block. The pinion by which the governor and oil pump are driven comes between the two as indicated. Views of the main bearing showing the cast-iron bush and the white-metal lining are given in Figs. 361 and 362, above. On the lower half of the bearing two gun-metal safety strips extend the full length of the brass, extending almost through

the white metal, being but 4 mils below the surface of the latter. These are intended to receive the shaft journal and prevent a "strip," should the white metal be melted out, owing to the oil supply failing through some accident. In one case an erector left a piece of waste in the oil pipe, but, though the white metal fused, the safety strips prevented serious damage being done. The two halves of the bearing are held together by four $\frac{1}{16}$-in. turned bolts. The $\frac{1}{4}$-in. grub screw shown in Fig. 362 prevents the bolt dropping when the upper half of the bearing is disconnected.

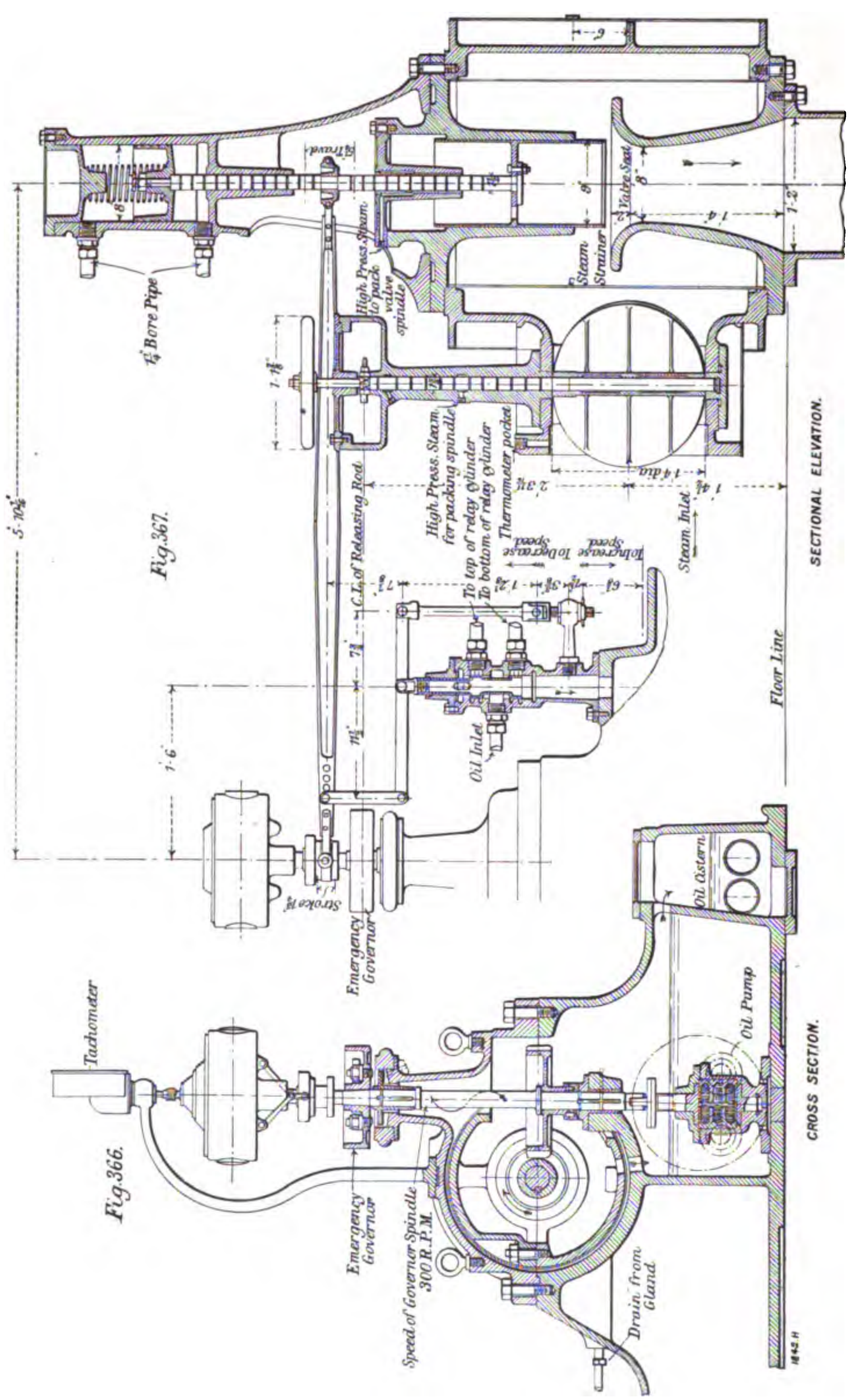
The flexible coupling by which the drive is communicated to the generator is of the claw type, and is illustrated in detail in



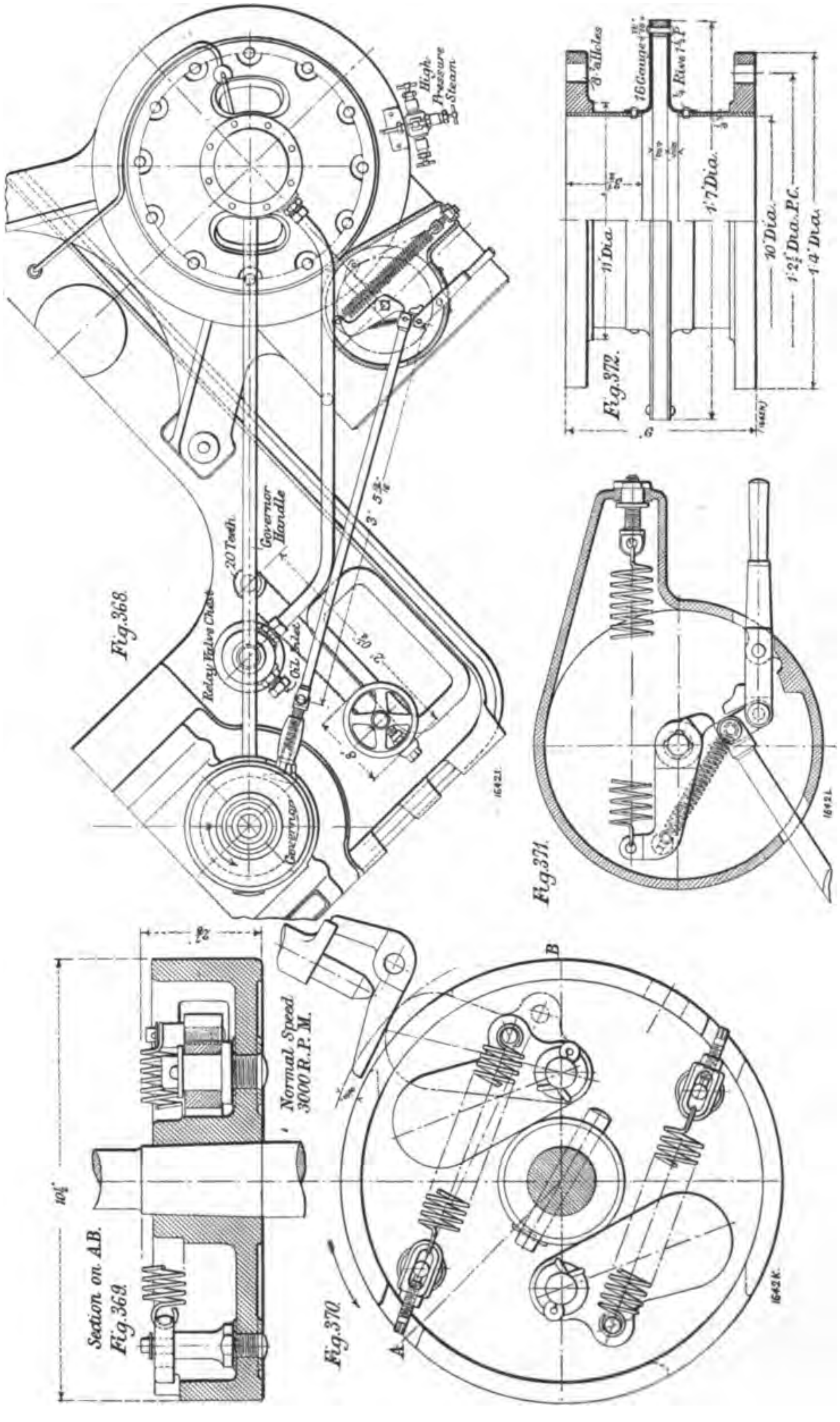
Figs. 363 to 365. Flexible Coupling; Brush-Parsons Turbine.

Figs. 363, 364, and 365. Each half of it is bored taper to fit a cone on the shaft, and is held in position by a fine-threaded (eleven per inch) nut, securely locked in position by plates fitting into a slot machined in the face of the half coupling and turned up over two of the flats of the nut, after the latter has been screwed home. A good view of this coupling complete and in position is given in Fig. 363.

The designed speed of the turbine is 2000 revolutions per minute, and the governor fitted can maintain this within a limit of $2\frac{1}{2}$ per cent., provision being also made for speeding it up or down to suit the voltage desired at the generator terminals. The turbine drives the governor spindle through spiral gearing. This



Figs. 366 and 367. Governor Gear for Brush-Parsons Exhaust-Steam Turbine.



Figs. 368 to 371. Details of Governor Gear; Brush-Parsons Exhaust-Steam Turbine.

Fig. 372. Steam Pipe Coupling.

spindle at its lower end operates the oil pump, which is of the positive rotary type, and carries also the emergency governor. Sections showing the details of the governor gear are represented by Figs. 366 and 367, page 290. The main governor is of the oil-relay type, taking its supply from the lubricating service. The governor valve is shown to the right of Fig. 367, and consists of a cylindrical shell, which is raised or lowered from or to its seat. The latter is near the throat of a convergent-divergent pipe, an arrangement which permits of a relatively light valve controlling with little frictional loss the flow of a very large volume of steam. The velocity of the latter at the point where it is regulated is high, and thus a small area can pass a large volume. The piston of the relay cylinder has, it will be seen, a spring above it which forces down the valve shutting off the steam should the oil pressure fail, a circumstance which would endanger the bearings of the turbine. The valves controlling the supply of the oil to the relay valve are shown a little to the right of the governor, Fig. 367. As the governor sleeve rises it carries up with it one end of a long aluminium lever extending from the sliding collar to the valve spindle. The relay valve is connected to this by the linkwork shown. If the speed increases the relay valve is raised, admitting oil pressure to the top of the relay cylinder, and thus forcing down the governor valve. The latter carries down with it the aluminium lever, and in so doing restores the relay valve to its normal position.

The emergency governor is of very simple construction, its details being clearly set forth in Figs. 369 to 371. It consists of a couple of pivoted weights which are held by springs tight against a central boss until the speed of the turbine exceeds the normal by 10 per cent. The centrifugal force then overcomes the tension of the springs, and increasing rapidly as the centre of gravity of the pivoted weights gets further from the centre of rotation, these weights practically instantaneously move up against their stops, in which case they protrude (as indicated by the dotted line in Fig. 370) through the wall of the casing. In this position they strike the tail of a pawl and release a rod which had previously held the emergency governor valve open. This valve is of the ordinary throttle type, as represented in Fig. 367 to the left of the main governor valve. Normally this valve is held open against the

tension of a strong spring by the rod just mentioned, which is, moreover, clearly shown in position in Fig. 368; but when this rod is released by the emergency governor, the spring instantaneously closes the valve and shuts off the steam from the turbine. After the emergency governor has acted, the valve requires to be opened by hand, which is effected by pulling back the hand lever shown in Fig. 371, page 291.

On trial at the builders' works the following figures as to steam consumption were recorded:—

TABLE XVIII.—STEAM-CONSUMPTION TESTS ON 600-KILOWATT EXHAUST-TURBINE SET.

Load.		Water per Hour, Pounds.		Steam at Stop Valve.		Steam Pressure below Governor Valve.	Barometer.	Vacuum.
Fraction of Normal.	Kilowatt.	Total.	Per Kilowatt.	Gauge Pressure. Pounds per Square Inch.	Temperature. Deg. Fahr.	Pounds Absolute.	Inches of Mercury.	Inches of Mercury.
½	306	10,770	35.2	− .610	229	9.5	29.85	28.82
¾	448	14,076	31.4	− .245	241	12.4	29.85	28.64
Full	596	18,876	31.7	− .170	256	15	29.85	28.28
,,	602	19,569	32.5	+ .8	222	15.5	30.28	28.41
,,	603	18,910	31.35	+ .4	253	15.4	30.28	28.41

The variant of the reaction steam turbine, constructed by Messrs. Willans and Robinson, differs in many details from the foregoing.

A longitudinal section, an end view, and a plan of a Willans-Parsons turbine, are given in Figs. 373 to 375, Plate XVIII. The steam inlet is at A, and the steam flows through the blades to the right, finally escaping to the condenser at C. As the pressure continuously diminishes from A to C, there is an end thrust on the rotor tending to make it move to the right. In the turbine illustrated, Messrs. Willans and Robinson have adopted the Fullagar system of balancing. The turbine, as shown, has only two dummy pistons at the high-pressure end of the turbine and one small one at the low-pressure end. The high-pressure dummy is lettered B in Fig. 373, whilst D denotes the intermediate dummy, and E the dummy at the low-pressure end, which, it will be seen, is quite

small in diameter. The back of the high-pressure dummy is connected by the pipe H with the exhaust from the high-pressure blades, and hence this dummy balances completely the thrust on the high-pressure blading. The back of the intermediate dummy is, however, connected direct to the condenser branch, and as a consequence the intermediate section of the turbine is overbalanced. This surplus is utilised to balance partially the thrust on the low-pressure blades, the residue being equilibrated by steam passing through the holes F and G to the back of the low-pressure dummy E.

As shown in Fig. 373, each half of the casing is in three lengths, registered together by a tongue and groove turned at each joint. The underlying motive of the design of this portion of the turbine has been to utilise simple circular castings without ribs or cored passages, likely to give trouble from unequal expansion. Each portion of the casing being in a short length, all the machining on it is done within full sight of the workman. The practice is to plane up first the longitudinal joint, and bolt together the halves into a complete ring, which is then bored out and the ends faced on a vertical boring mill. Coincidence between the centre of the bore and the plane of the longitudinal joint is secured by careful work. All bolt holes in all flanges are drilled to jigs, so as to secure thorough interchangeability.

Two methods of constructing the rotors are illustrated. That shown in Fig. 381, page 295, represents the pattern which has been used for turbines rated at 2000 kilowatts or under, whilst in Fig. 382 is shown a pattern used for larger sizes. Taking the former first, the most noteworthy feature lies in the fact that the high-pressure end is made entirely in one piece.

At the low-pressure end of the rotor the shaft is separate from the main forging, as shown. It is secured to this forging by T bolts, let into a slot turned in the low-pressure end of the drum, as shown separately in Fig. 380. When the nuts have been tightened down, the projecting ends of these bolts are riveted over, making them thoroughly secure. When, as in Fig. 382, the rotor is distinct from the high-pressure end shaft, special precautions are always necessary to prevent differential expansion at the joint. The plan followed is clearly shown in Fig. 382, which represents the drum

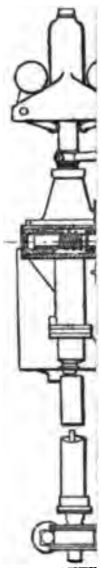


Fig. 374.

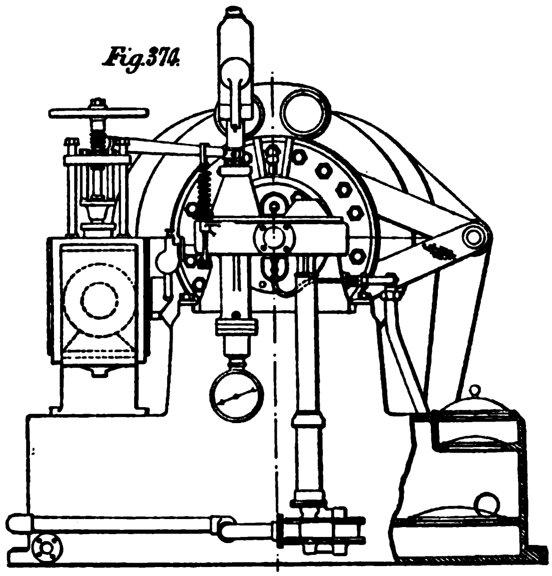


Fig. 375

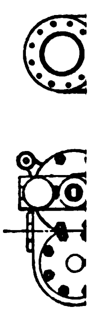


Fig. 376.

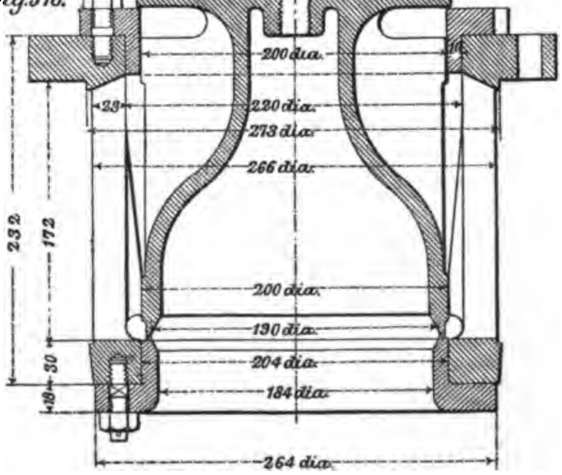
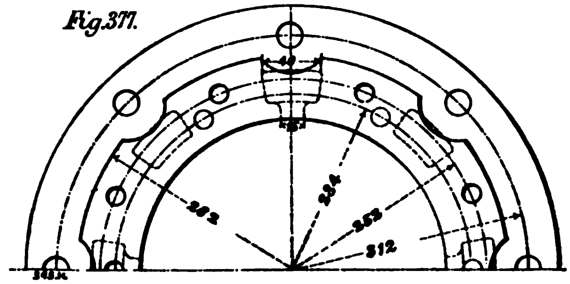
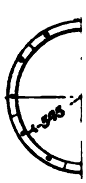


Fig. 377.





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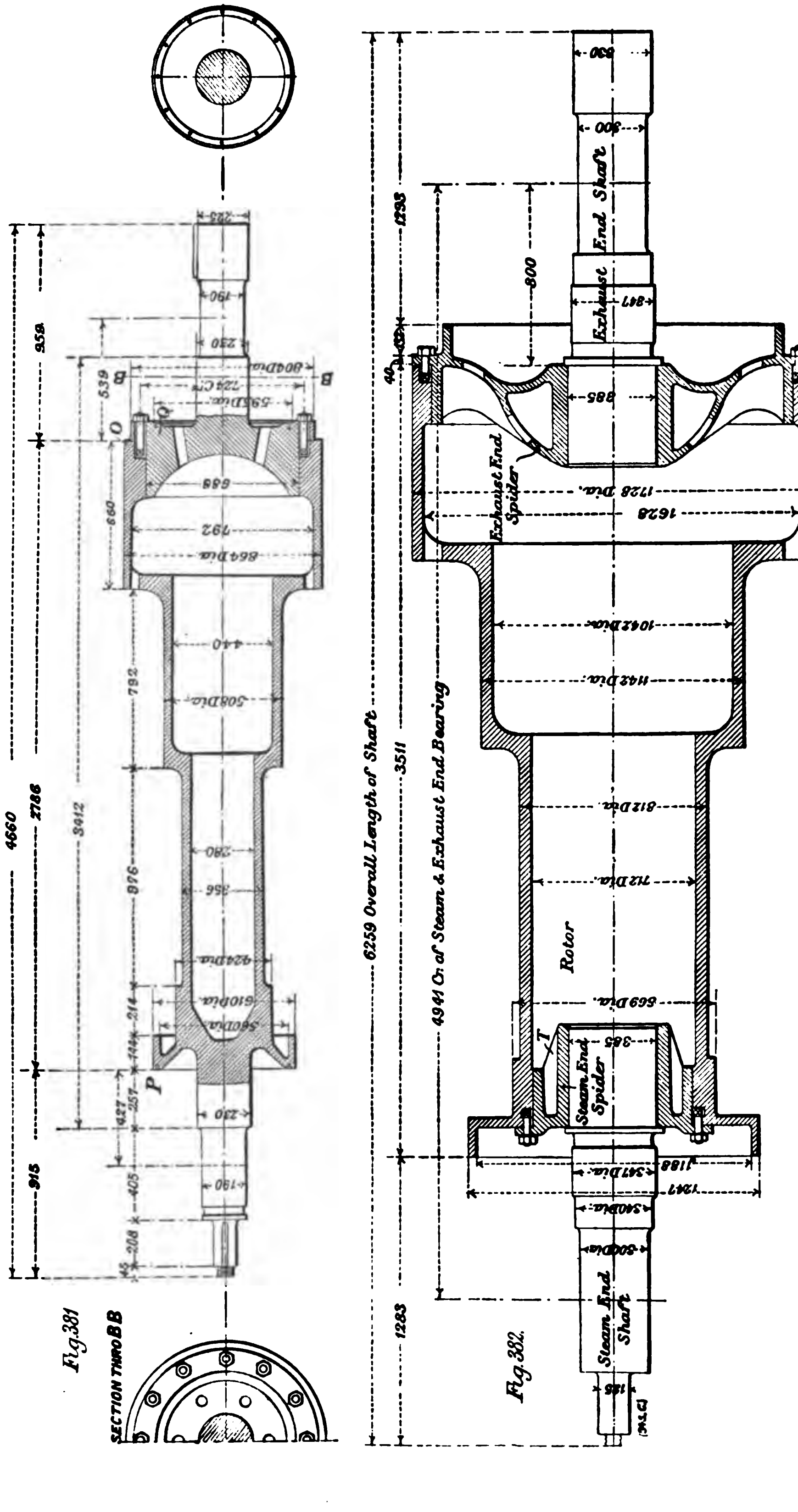


Figs. 381 and 382. Rotors for Willans-Parsons Turbines.

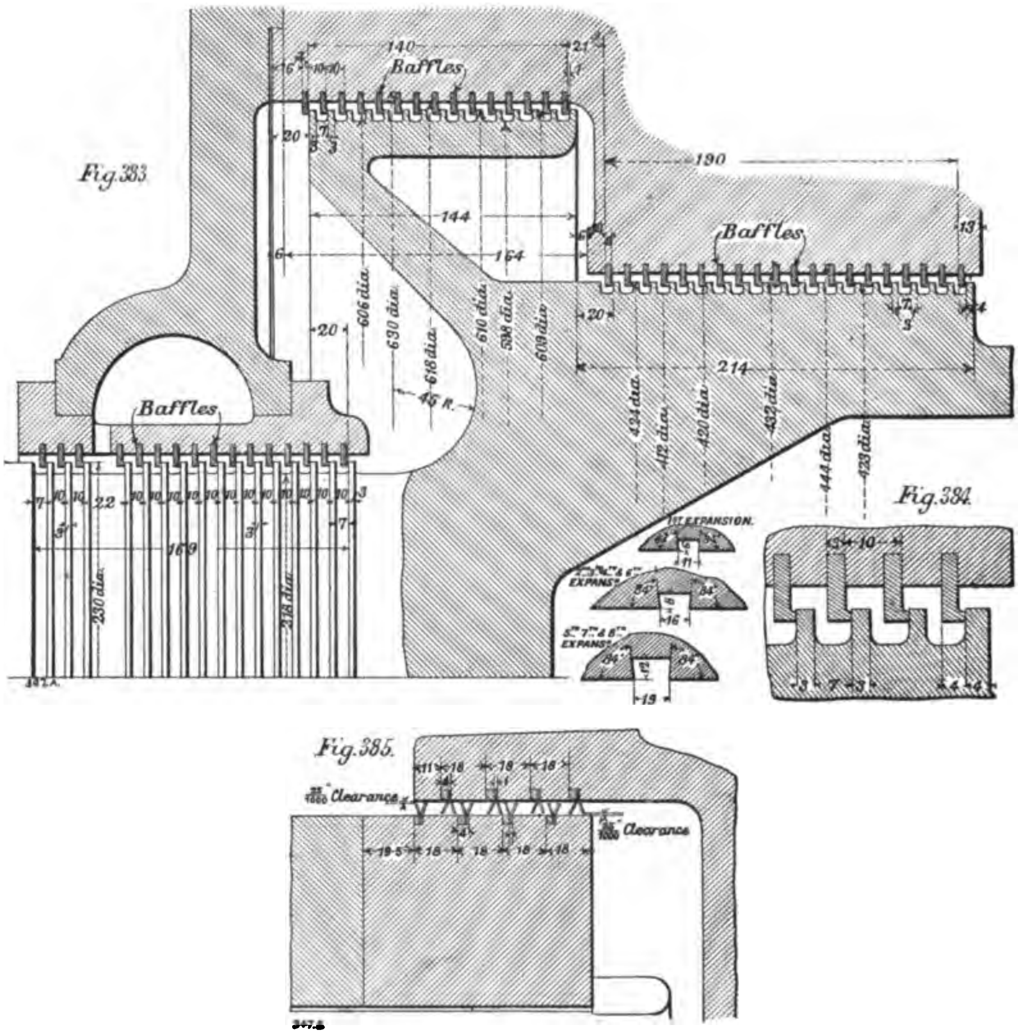
of a 3000-kilowatt machine. The stub end of the shaft is let into a steel casting, which is cored out at T, T, as indicated. The portion of the spider into which this stub end of the shaft is forced is exposed to the temperature of low-pressure steam only, whilst the outer portion, which does come in contact with the often highly heated drum, is secured to the latter by T bolts let into a slot, as already explained, in dealing with the low-pressure end of the 2000-kilowatt rotor.

The spider for the low-pressure end of the 3000-kilowatt rotor illustrated is constructed on the lines of the American car-wheel, with the view of eliminating any internal strains which might arise in the process of casting. Like the steam end spider, it is a steel casting.

Details of the labyrinth packings are illustrated in Figs. 383 to 385. At the low-pressure end provision must be made for axial movement, and thus the labyrinth is of the axial-flow type. It consists, it will be seen, of a number of fins on the casing, which come almost in contact with the shaft, and an equal number on the shaft almost in contact with the casing. The outer end of each fin is split. The clearance between the shaft and the fins must, it is found, be larger than where radial-flow packings are used, and for 8-in. shafts may be 30 mils. This type of packing is used not only for the glands, but also for the low-pressure dummy. The function of the glands, which are shown at V and W, Fig. 373, Plate XVIII., is, of course, not to prevent the leakage of steam out of the casing, but to stop the entrance of air.

The main bearings of the turbine are arranged close to the glands. The construction of these bearings is shown in detail in Figs. 386 to 389, page 298. The bearing surface is of white metal, hammered into cast-iron shells. The two half shells are bolted together by turned bolts, and have secured to them, at top and bottom and at the sides, four pads of cast iron, which fit into the recesses shown at R, Q, S, and T, Fig. 388, and are secured to the shell by countersunk screws. Packing pieces of steel of various thicknesses are placed between these pads and the shell. The pads are then turned to a spherical surface, to correspond with a similar spherical surface bored in the turbine framework to receive them. The object of fitting packing pieces under the pads is to provide a method of

adjusting the bearings true to the bore of the casing when the turbine is being erected. By taking out one of the upper packing pieces, and inserting it amongst the lower, the effective centre of the bearing is raised, whilst a sideways adjustment can be effected

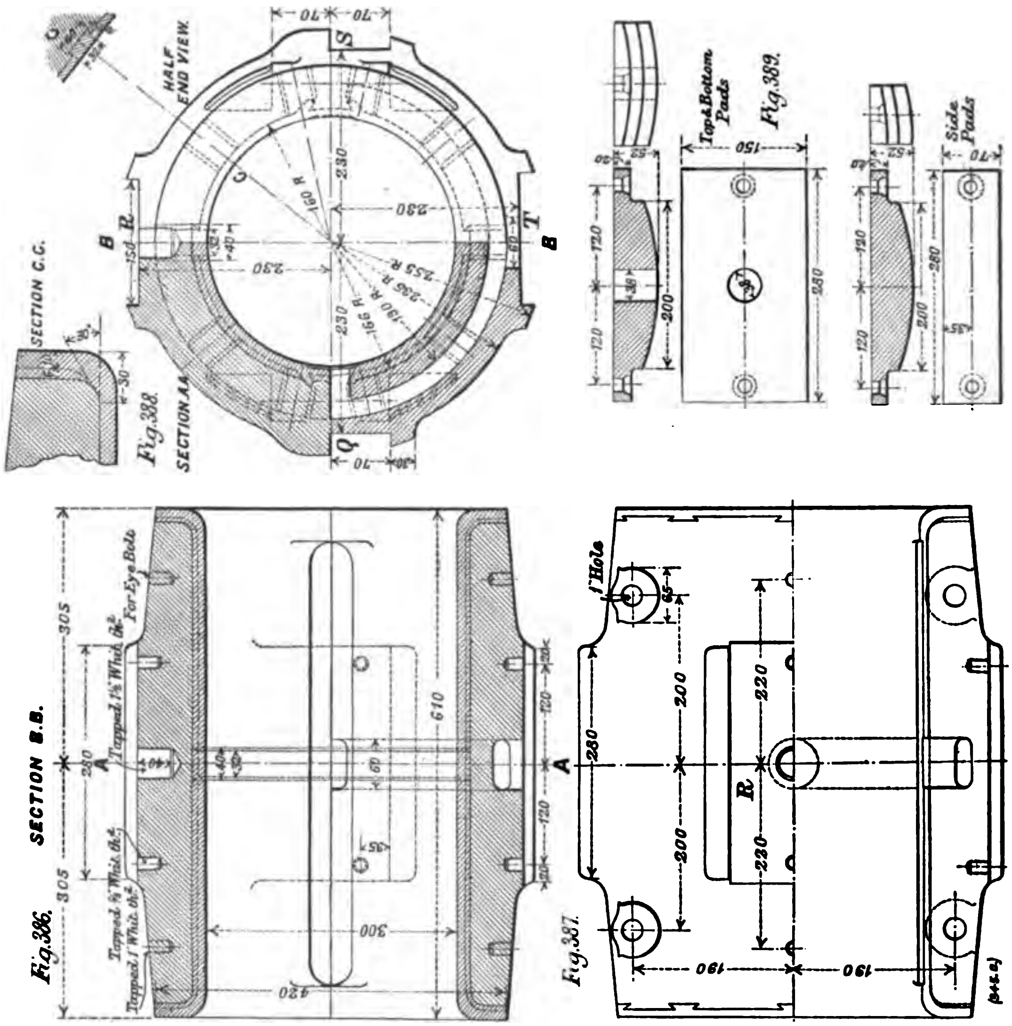


Figs. 383 to 385 Dummy and Gland Packings for Willans-Parsons Turbines.

in similar fashion by thickening and thinning the packings under the side pads.

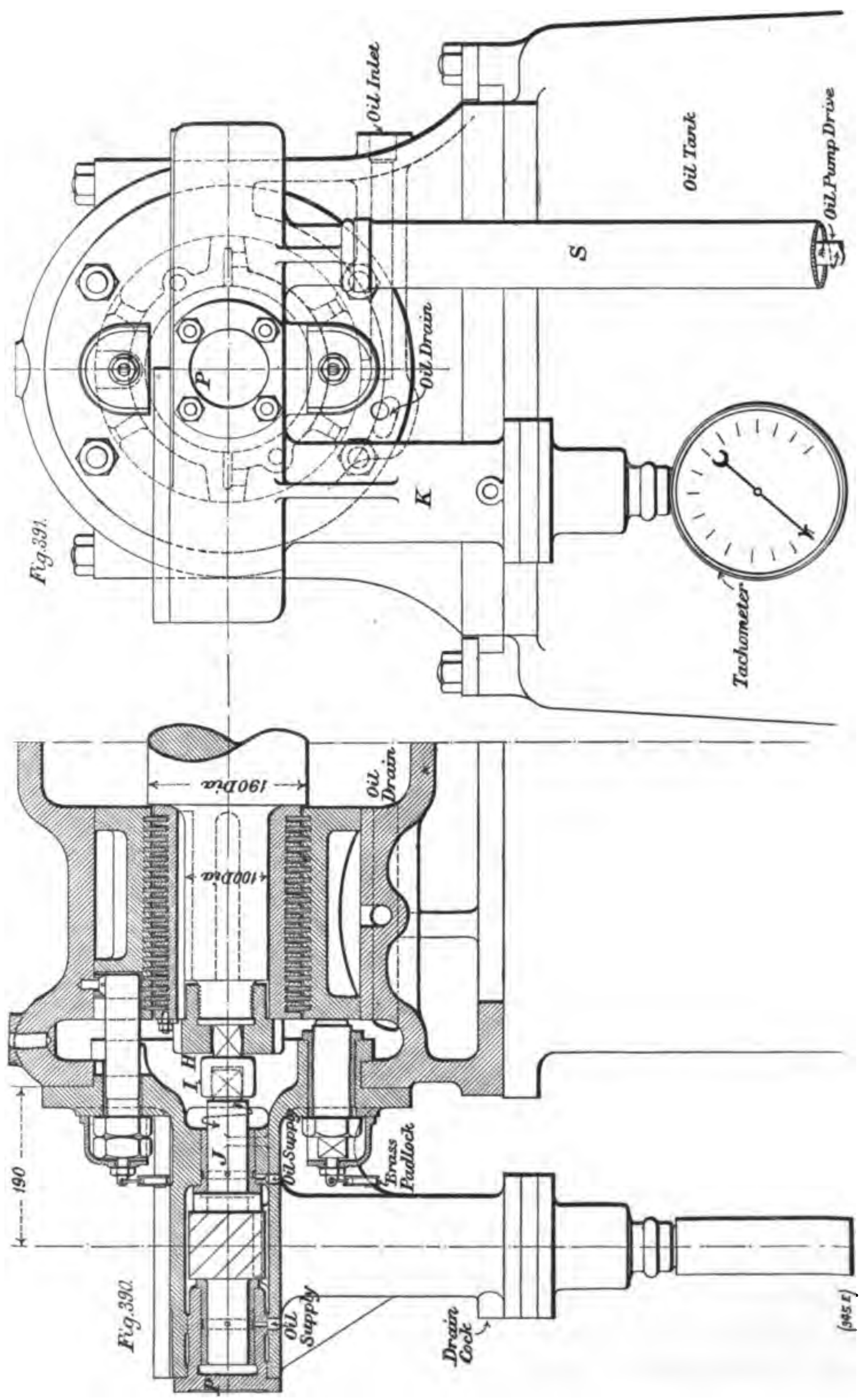
The method by which oil is supplied to the bearings is best seen at the low-pressure bearing in Fig. 373, Plate XVIII. Oil is forced up a hole provided for this purpose in the bottom pad, and thus enters a channel cast in the shell, as shown in Figs. 386 and 387.

This communicates at each horizontal diameter with a long shallow slot formed in the white metal, and it is from this channel that the oil is distributed to the upper and lower halves of the bearing. It flows axially along the shaft, and escapes from the bearing at



Figs. 386 to 389. Main Bearing Brasses for Willans-Parsons Turbines.

each end. Oil flirts are provided to prevent the creeping of the oil along the shaft to the exterior of the casing. The oil, after leaving the bearing, falls into a receiver below, and thence by piping to the cooler, which is shown at L, Fig. 373. When cooled and strained, the oil is returned to the bearings by a pump driven direct from the turbine shaft by worm gearing.



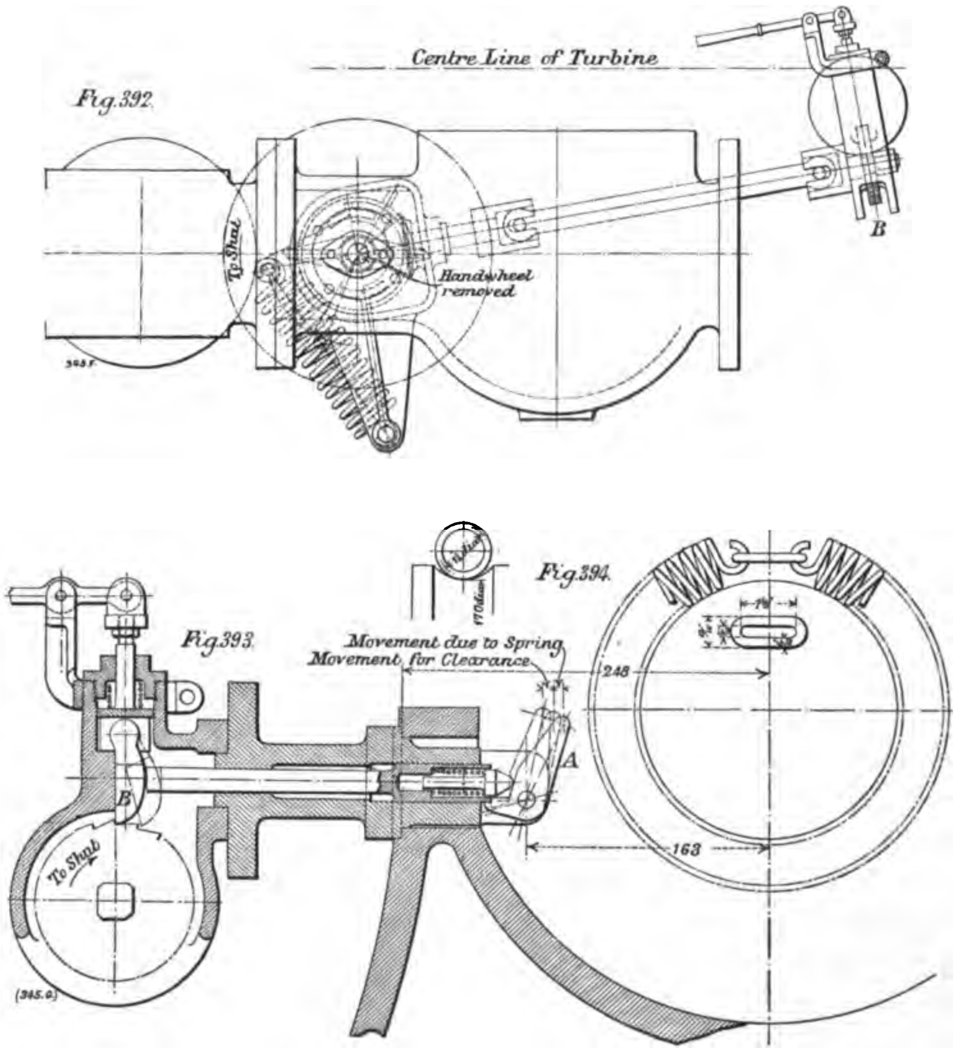
Figs. 390 and 391. Tachometer and Oil Pump Drive; Willans-Parsons Turbines.

To the immediate left of the bearing at the high-pressure end (Fig. 373) will be seen the emergency governor M, which will be described in detail later on. Beyond this comes the thrust block, shown to an enlarged scale in Fig. 390, page 299. This fits into a recess bored in the turbine framing; its position is adjustable axially, the upper half by the T-headed bolts shown, and the lower half by a fine thread set bolt. In making the adjustment, the rotor is pulled to the left by the T bolts until the dummy packing rings comes into actual contact. The micrometer set bolt is then screwed in to bring the lower block in contact with the collars on the shaft. This done, the upper half of the block is slacked back, and the lower half adjusted so as to give the desired clearance—10 mils or so. The upper half is finally brought up again by the T bolts, so as to give the thrust collars about 5 mils play in the slots. The whole is then locked in position. The oil supply is pumped in from below and escapes at the top. The thrust collars, it will be seen, are cut in a sleeve secured to the shaft by a left-handed nut H (Fig. 390), which is shown in detail in Fig. 379, Plate XVIII. There is a square hole in the end of this nut, into which fits the rounded end of a junction piece I (Fig. 390), which drives the governor and oil-pump-actuating shaft J. This shaft carries a multiple-threaded worm, which on the one side gears with a wheel on the governor and tachometer spindle at K (Fig. 391), and on the other with the oil-pump spindle at S. The worm and its shaft can be removed bodily, without disturbing any other portion of the mechanism, by taken off the cover P (Figs. 390 and 391). The thrust of the worm is taken on a Hoffman ball bearing.

The governing of the turbine is effected by an ordinary throttle valve and not by a relay. The main governor valve is shown in detail in Figs. 376 and 377, Plate XVIII. It is a double-beat equilibrium valve of cast iron, chilled at the seats, and finished by grinding. The seats are also of cast iron, and the lower one is secured by square-bodied studs to the valve cage. The square bodies of the studs fit into square holes in the seat rings, which effectually prevent the loosening of the studs by the vibration set up. The governor, which is well shown in the general view, Fig. 373, lifts or lowers this valve, as needed, by the lever shown. The plan view shows how the stop valve, governor valves, and by-pass valve of the turbine are arranged.

Starting from the left, the stop valve is at W, the throttle valve at X, and the by-pass valve at Z, the emergency governor valve being between the two latter.

The emergency governor is of a very simple and ingenious type.



Figs. 392 to 394. Emergency Governor and Gear; Willans-Parsons Turbine.

It consists of a coil spring fitting the shaft, as indicated at M, Fig. 373, and shown on a larger scale in Figs. 393 and 394, above. When the speed becomes excessive, the centrifugal force on this spring becomes greater than its tension, and it expands. Not being in balance, as is evident from Fig. 393, it continues to revolve with the shaft, and, its expansion increasing, it finally strikes the trigger

at A, Fig. 393. This unlocks the pawl B, which engages with the teeth of a ratchet wheel geared to the spindle of a throttle valve, which immediately closes under the action of a spring. The connection between the ratchet wheel and the valve is made by means of a Cardan shaft, as shown in Fig. 392, page 301.

The blading is fixed on machine-divided foundation strips, which are held in place by caulking rings.

The shrouding rings are of channel section, and are drawn to shape. They are bent to a half-circle, and are punched to receive the tips of the blades. The operation of punching causes the strip to elongate by about $\frac{1}{40}$ in. per foot, and to provide for this it is clamped to the cylinder of the punching machine behind the punch only. The portion yet to be perforated is held but loosely, and hence all the expansion takes place in this direction, and the accuracy of the spacing remains accordingly unaffected.

It is usual in steam-turbine practice to fit an expansion joint between the turbine and the condenser, in order to avoid strains on the turbine casing from the expansion of the condenser by heat. When such a joint is used, however, there is a large unbalanced load to be carried by the turbine casing whenever there is a vacuum in the condenser, and to avoid this the plan has been adopted in this instance of mounting the condenser on springs strong enough to take the whole of its own weight and of the water flowing through it. The condenser can then be bolted direct to the exhaust branch, and when hot expands downwards by compressing the springs. This additional compression does, of course, give rise to a slight upward pressure against the turbine casing, which is, however, very small in comparison with the downward force it has to withstand when an expansion joint is used.

CHAPTER XXVII.

GENERAL NOTES ON TURBINES OF THE PARSONS TYPE.

THE casing of steam turbines of the Parsons type is generally of cast iron, but with the high superheats the high-pressure end is best made a steel casting. Steel is advisable in such cases because cast iron "grows," that is to say, permanently increases in volume, when exposed for a prolonged period to a very high temperature. The necessity for the adoption of steel is likely to become more imperative with the continuous increase in superheats. In French admiralty practice high-pressure marine turbines are tested to 90 lb. more than the designed working pressure, and under this a stress on the cast iron of 2 tons per sq. in. has been permitted. The low-pressure turbines are subjected to a test load of twice the working pressure, and the stress on the metal in that case is of the order of $\frac{9}{10}$ tons per sq. in. The essential feature to be borne in mind in designing the casing is to secure an absence of distortion. Equalising ports, connecting the dummies with the discharge end of the sections they balance, should be separate pipes, and not cast in the turbine casing. The latter should be well protected by a non-conducting jacket when at work, as experiment has shown that an uncovered casing "hogs" perceptibly when hot, owing to the lower part of the casing being cooled by convection currents more than the upper. If the highest economy is to be secured the casing is best divided into a separate high and low-pressure portion. The dummy clearances necessary are largely dependent on the maximum diameter of the casing, and since high-pressure steam is very dense, it is very important to keep these clearances small at the high-pressure end. Dummy clearances are commonly about 5 mils per ft. of the maximum dummy diameter. The flange at the horizontal joint of the turbine casing can hardly be too thick. The ideal casing would be a joint-

less ring in which every section which was originally radial remains so under strain. With a casing made in halves, however, and having thin joint flanges, the joint faces tend to open out under pressure, and this tendency is kept in check solely by the stiffness of the flange. The latter may therefore well be made $1\frac{1}{2}$ to $1\frac{3}{4}$ times the shell thickness. The bolts should be brought up as close as possible to the inside edge of the joint, and it is therefore not uncommon to use here special nuts and bolts with small heads in place of standard patterns.

With modern surface condensers vacua in the neighbourhood of 29 in. may be secured in temperate climates during several months of the year. To utilise such efficiently the low-pressure end of the turbine is sometimes made on the double-flow system.

Where high superheats are to be employed the belt through which the steam enters the casing is best made so as to lie equally on each side of the port. If made so that there is a large overhang on the one side there is a danger that the greater expansibility by heat of the brass blades and caulking pieces may deform the casing, causing the blades to foul. The horizontal joint for the casing may be made with boiled oil and plumbago, or with magnesite painted on.

The rotor was at one time built up of steel castings, but forgings are now most commonly used. These have the advantage of being everywhere of uniform density, and thus are more easily balanced. Some firms, however, now offer to supply steel castings more than satisfying the Admiralty specifications for steel forgings, so that the use of castings may be revived. The table opposite, taken from the paper contributed to the proceedings of the Junior Institution of Engineers by Mr. J. M. Newton, B.Sc., shows the character of the steel forgings commonly used.

In the early days of high superheats there were a number of cases of blades stripping owing to differential expansion of the body of the rotor relatively to the shaft socketed into it. The trouble arose not because the superheat was high but because it was variable. If the load on a generator suddenly fell off and came on again the temperature of the steam was liable to range through 200 deg. to 300 deg., and to such conditions the earlier forms of stub-end attachments proved unequal. Two different methods of surmounting this

difficulty have been described in the preceding Chapter. In the case of turbines for generator driving, there is an absolute absence of corrosion so far as the superheat extends, but beyond that there may be a little pitting of the drum. Marine turbines have at times shown signs of corrosion in the interior of the drums, which has been checked by a coat of whitewash, and by arranging for efficient drainage.

TABLE XIX.—STEEL FOR TURBINE ROTORS.

Specimen.				Stress.				
No.	Material.	Diameter.	Distance between Gauge Points.	Elastic, Tons per Square Inch.	Ultimate, Tons per Square Inch.	Elongation.	Reduction of Area.	Fracture and Remarks.
		in.	in.			p. c.	p. c.	
1	Nickel steel	0.780	2	30.2	44.4	30	53.8	Silky fibrous
2	,,	0.798	2	29.5	40.8	30	55.8	,, ,,
3	,,	0.798	2	29.8	40.4	34.5	59.2	,, ,,
4	Mild steel	0.798	2	22.2	40.5	25	35.6	,, ,,
5	,,	0.796	2	24.4	39	30.5	54	,, ,,
6	,,	0.787	2	26.5	38.6	28	46	,, ,,
7	,,	0.800	2	25.8	40	26	36	,, ,,
8	,,	0.800	2	26.2	36	34	56	,, ,,
9	,,	0.798	2	17.2	35.4	16	20.8	20 per cent. fibrous, 80 per cent. finely granular (taken from top of ingot)
10	,,	0.798	2	15.8	32.9	30	47.2	Silky fibrous (taken from bottom of ingot)

Specimens 9 and 10 failed to meet the requirements and were rejected.

In land practice the blading is commonly made to three diameters, as represented in Fig. 382, page 295, *ante*. The turbine is thus divided into a high-pressure, an intermediate, and a low-pressure section. The two former commonly do each about one quarter the total work, the remainder being left to the low-pressure section. The drum diameters in the different sections commonly increase in the ratio of $1 : \sqrt{2} : 2$.

With powers which are not large compared with the speed of

rotation this system of construction is quite satisfactory, but if it is intended to build a 2000 or 3000-kw. machine to run at 3000 revolutions per minute it is preferable to make the rotor in four diameters, as in the case of the Brush turbine described, page 269, *ante*. The desirability of this is due to the enormous volume of steam at the exhaust end, and its consequent high velocity if the passage way be restricted. If r be the residual velocity of the steam as discharged from the last row of blades, the loss by "carry-over" to the exhaust is $\left(\frac{r}{224}\right)^2$ B.Th.U. per lb. of steam passed. The blade height cannot well exceed $\frac{1}{5}$ the mean diameter, so that to pass a very large volume of steam, as is necessary when the output is great, a large diameter is needed at the low-pressure end if the final velocity of discharge is to be moderate. In extreme cases the exhaust end may be constructed as a disc, carrying a single row of blades, in which case it would be possible to construct a reaction turbine of as much as 8000 kw. to run at 3000 revolutions per minute, with a loss by carry-over to the exhaust of not more than some 5 per cent. of the total energy supplied. An alternative to the plan of making the drum in four diameters is to construct the low-pressure end on the double-flow system.

In marine practice, when three shafts are used there is one high-pressure turbine doing one-third the total work, and two low-pressure turbines, arranged "in parallel," doing the remainder. In the case of four-shaft ships there is usually, on each side of the vessel, a high-pressure turbine driving a wing shaft and developing about one quarter the total power, which is in series with a low-pressure turbine of equal power, which drives one of the centre shafts. The low-pressure drum, whichever arrangement is used, is usually about $\sqrt{2}$ times as great in diameter as the high-pressure drum.

The Blading.

Rules for fixing the number and proportions of the blading are given in Chapter VII. The blades used are a "form of least resistance." Good proportions for normal blades have already been given in Fig. 14, page 23. They are commonly pitched closer on the drum than in the casing, but the opening in standard practice is always one-third the pitch. Some builders, however, "gauge" their

blades so as to keep the speed of the steam constant throughout each group or expansion. In marine work gauged blades are not now used (1912).

The angle of discharge from $\frac{3}{8}$-in. normal blades gauged with an opening equal to one-third pitch has been measured by Messrs. W. Chilton and J. M. Newton, as already described on page 23. So called semi-wing blades are merely normal blades having special spacers, which give them a discharge angle of 28 deg. to 30 deg.

A good form of wing blade is represented in Fig. 15, page 23. The discharge angle is about 40 deg., but in cases is made as much as 50 deg. A table of standard blades and caulking pieces is given below :—

TABLE XX.—STANDARD PARSONS BLADES AND CAULKING PIECES.

Blades.			Caulking Pieces.									
			Casing.					Drum.				
Section No.	Weight per Foot Run.	Usual Limits of Length.	Section No.	Number Required per Inch Run.	Weight per Foot Run.	Grooves. Width.	Grooves. Depth.	Section No.	Number Required per Inch Run.	Weight per Foot Run.	Grooves. Width.	Grooves. Depth.
						in.	in.				in.	in.
120 B	0.041	Up to $\frac{3}{4}$ in.	120 C	5.8	0.106	0.243	$\frac{1}{4}$	120 C	5.8	0.106	0.243	$\frac{1}{4}$
130 B	0.089	$\frac{3}{4}$ in. to 4 ,,	130 C	3.7	0.264	0.364	$\frac{1}{4}$	130 S	5.2	0.159	$\frac{23}{64}$	$\frac{1}{4}$
130 B	0.089	$\frac{3}{4}$,, ,, 4 ,,	131 C	3.3	0.315	0.415	$\frac{1}{4}$	131 S	3.7	0.300	$\frac{27}{64}$	$\frac{1}{4}$
132 B	0.077	$\frac{3}{4}$,, ,, 4 ,,	132 C	3.2	0.400	0.406	$\frac{1}{4}$	132 S	3.6	0.320	0.400	$\frac{1}{4}$
240 B	0.180	4 ,, ,, 8 ,,	240 C	2.75	0.450	0.510	$\frac{3}{8}$	240 S	3.4	0.400	0.510	$\frac{3}{8}$
240 B	0.180	4 ,, ,, 8 ,,	241 C	2.54	0.643	0.540	$\frac{3}{8}$	241 S	3.2	0.450	0.540	$\frac{3}{8}$
250 B	0.246	8 ,, ,, 12 ,,	250 C	2.2	0.783	0.610	$\frac{3}{8}$	250 S	2.7	0.600	0.610	$\frac{1}{2}$
250 B	0.246	8 ,, ,, 12 ,,	251 C	2.1	0.917	0.661	$\frac{3}{8}$	251 S	2.6	0.670	0.661	$\frac{1}{2}$
252 B	0.237	8 ,, ,, 12 ,,	252 C	1.96	1.080	0.716	$\frac{3}{8}$	252 S	2.44	0.790	0.716	$\frac{1}{2}$

The section numbers are arranged on the following plan:— The last digit in each case represents the class of blade, whether normal, semi-wing, or wing, each being denoted respectively by 0, 1, or 2. The semi-wing blades are normal blades with special spacers, as shown in the table. The second digit gives the width of the blade in eighths of an inch, measured parallel to the turbine axis. Thus 130 B means a normal blade $\frac{3}{8}$ in. wide, for which the casing spacer is 130 C, and the drum or spindle spacer 130 S. If used as a semi-wing blade, the spacers would be 131 C and 131 S respectively.

Mr. E. M. Speakman has given the following table, showing the axial pitch used with different sizes of blade:—

TABLE XXI.—AXIAL PITCHES FOR PARSONS BLADES.

	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.
Height	1	2	3	4	6	8	10	12	15	18	21	24	30
Width	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	1	1	$1\frac{1}{8}$	$1\frac{1}{4}$
Axial pitch	$1\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{5}{8}$	$1\frac{3}{4}$	$2\frac{1}{8}$	$2\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{5}{8}$	$3\frac{1}{8}$	$3\frac{1}{4}$	$3\frac{5}{8}$	4
Axial clearance (c) ...	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$

With $\frac{1}{4}$-in. blades an axial pitch of $\frac{3}{4}$ in. is used. The axial pitch is, it will be seen, large, but this is advisable in order to avoid the risk of distorting the drum by the operation of caulking, which would ensue were the grooves close together. A large axial clearance moreover eliminates any risk of the blades fouling in opening up the turbine.

The steam speeds being low in the case of Parsons turbines, blades of brass and copper have proved very satisfactory. The copper blades are used for such portions of the turbine as are likely to be subjected to high superheats, the brass blades being adopted elsewhere.

Mr. J. M. Newton, "Junior Institution of Engineers," vol. xx., states that this brass consists of 63 per cent. copper and 37 per cent. zinc, and has the following mechanical properties:—

TABLE XXII.—MECHANICAL PROPERTIES OF BRASS FOR BLADING.

Specimen.	Stress.		Elongation per Cent. in 2 In.	Reduction of Area per Cent.
	Elastic, Tons per Square Inch.	Ultimate, Tons per Square Inch.		
Cast rod 1 in. in diameter	16	21.6	44	45
Drawn blade, area of cross section = 0.0105 sq. in. ...	20.50	23	17	60

The blades have to take a bending moment as well as a centrifugal pull. This bending moment is in part due to the drive of the blades and in part due to the pressure drop. Each of these component bending moments is easily calculated, and from them the resultant moment in the direction of the least resistance of

the blades. The ultimate resistance to bending is approximately equal to $1.2\ w^3$ in.-lb. in the case of normal blades, and to $\frac{w^3}{1.54}$ in.-lb. in the case of wing blades. In this formula, w stands for the nominal width of the blade taken in $\frac{1}{8}$ of an inch. Thus a $\frac{3}{8}$ in. normal blade will fail under a bending moment of $1.2 \times 3^3 = 32.4$ in.-lb.

The material of these blades is "work hardened," and it is probable that if the blades were subjected to prolonged high temperatures their resistance would be diminished. It is, however, only the low-pressure blades which are, in normal conditions, subjected to serious centrifugal or bending stresses.

The blades of reaction turbines, particularly at the low-pressure end in the case of electric generator turbines, occasionally "silt up" with a soft deposit of lime carried over from the boiler. This is, however, very slightly adherent, and can easily be rubbed off, and the metal below is then found to be clean and fresh. This lime also seems to attack slightly the inlet edges of the blades near the tips of the rotor and near the roots of the casing blades. The action is exceedingly slight, after three years' running being barely visible to the eye; but if the finger be rubbed along the last half inch of the low-pressure rotor blades, the edge, when much lime has been carried over by the steam, has the feel of a very fine saw. Both faces of the blades appear untouched, but the solid lime being flung outwards by the centrifugal force has "sand-blasted" the edges of the rotor blades near the tips, and of the stator blades near the roots. With the higher velocities usual in impulse turbines the action under similar conditions is much more severe, and in one case, in which blades of a similar brass to that commonly adopted in reaction turbines were used, in a velocity-compounded impulse turbine working with exhaust steam, they were cut away in three weeks. It is therefore advisable to use harder materials for impulse blading than for those which prove best with the reaction type. Steel is mechanically strong, and resistant to erosion, but is liable to corrosion under unfavourable conditions. Hence, efforts are still being made to find a bronze which is hard enough to resist wear. Messrs. Melms and Pfenninger have used one consisting of copper 72 per cent., pure zinc 28 per cent., and another consisting of 85 per cent. of copper

and 15 per cent. of manganese. Up to 390 deg. Fahr. the former has a tensile strength of 34 to 36 tons per sq. in., which at 572 deg. Fahr. has fallen to 24.1 tons. The other, having a tensile strength of 41.3 tons per sq. in. when cold, is still good for 36.8 tons at 572 deg. Fahr., but it is not certain that this would be retained on prolonged heating.

The system of fixing the blades on foundation rings, as in Fig. 395, has the advantage of being cheap, of setting the blades automatically to the desired gauging, and when, as is usual, a shrouding is added at the top of the blades great mechanical strength is secured. Its disadvantages lie in the fact that wider

Fig. 395.

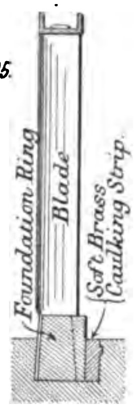
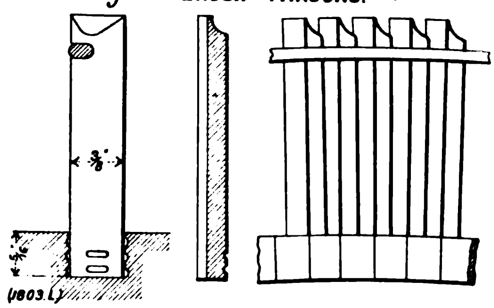


Fig. 396. BRUSH-PARSONS.



Figs. 395 and 396. Types of Blading.

grooves must be cut in the rotor, thus reducing its resistance to centrifugal forces. Further, where high superheats are used, it is advisable to make the foundation rings of steel, as there have been several instances of brass foundation rings working out in such cases, owing to differential expansion. Generally, moreover, the use of foundation rings involves the distortion of the blades near their roots, with a consequent reduction in their efficiency there, and when, as is often the case, the foundation rings project, there are losses due to the formation of eddies in the dead spaces. The system of caulking in the blades separately, as illustrated in Fig. 396, makes it possible to use narrower grooves in the rotor. At one time it suffered from the disadvantage that blading could not be commenced until the rotor and casing had been completely machined. The introduction of the "rosary" system of assembling

the blades has eliminated this drawback. The blades and distance pieces are now drilled and strung alternately on a soft iron wire. They are then transferred to cast-iron jigs, having grooves of the same depth and curvature as the rotor or casing, but 0.003 in. to 0.005 in. narrower. A view of a series of blades thus assembled is shown in position in the jig in Fig. 397. The whole series is driven up tight by the curved set shown. The ends of the wire are turned up to secure them from slacking back and then brazed, after which the projecting ends are filed off. The segment is then returned to the "former" and the blades gauged, the whole being finally secured by a steel wire, silver soldered into notches near the tips, as shown.

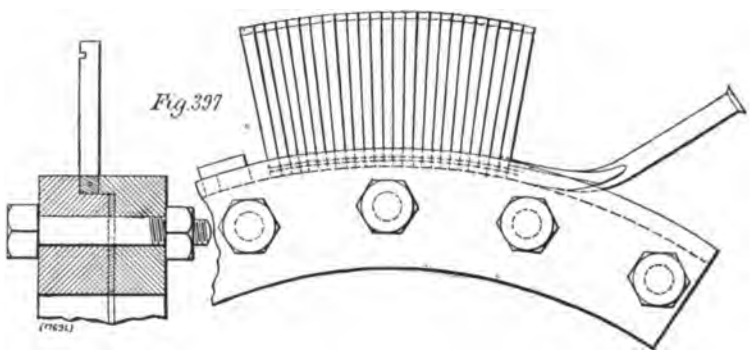


Fig. 397. The "Rosary" System of Assembling Blades.

Until the introduction of this system of assembling the blades in jigs, it was impossible to add this wire in the case of short blades, since the large mass of the rotor or casing conducted away the heat too quickly to permit of the requisite temperature being reached for the solder to flow. From six to eight of these segments constitute a complete ring of blades. They are prepared whilst the rotor and casing are being machined, and can then be transferred bodily to these, and finally caulked into place. The latter operation should be executed by conscientious workmen, and is, perhaps, best done by day work, since, if scamped, the defect is not likely to be discovered till after the turbine is put into service.

Both systems of construction have shown considerable security against blade stripping. The essential point, as developed by Sir C. A. Parsons, and adopted since by builders of every variety of turbine, is that where two portions of a turbine, in rapid relative

movement, are contiguous, one must be made as thin as possible. If both are stout and a touch occurs, the heat developed deforms the adjacent parts, and a serious breakdown is likely to occur. This fact was not realised with some of the earlier patterns of impulse turbines, with the result of wrecked diaphragms and bent or broken shafts. In the case of the blading of reaction turbines, the practice of thinning the blade tips has practically abolished the liability to strip from the touching of adjacent parts. As showing the efficiency of the device, the author saw, some years ago, a case where some of the groups of the turbine had blades with thinned and others with square ends. On opening up the turbine after a touch, the thinned blades were found to have simply worn down, whilst an intermediate group of square-ended blades had completely stripped.

Referring to Fig. 395, it will be seen that a similar principle is employed in the case of turbines having shrouded blades carried in foundation rings. The edges of the shrouding which may come in contact with the adjacent rotor or casing can only touch along thinned edges. Here, again, in the case of an old-type turbine in which contact had occurred through the high-pressure shaft coming loose through differential expansion, the thin ribs simply wore down, and there was no strip.

With a well-designed casing, the intended and actual clearances agree very well when the turbine is cold, but if the casing is unsymmetrical in form, or subject to mechanical restraint, there is no certainty that the actual clearances when at work will be the same as when cold. It is absolutely impossible to measure tip clearances with the turbine hot. Attempts have been made to do this by opening up the casing as soon as possible after the turbine is stopped, but this opening instantly gives rise to serious distortions owing to the inequality of the convection currents then established

It may, perhaps, be pointed out in passing, that in general a strip with a reaction turbine is less serious than it is with impulse turbines as usually constructed. A reaction turbine can be opened up and the broken blades removed, when it may be set to work again, the loss of a whole expansion hardly affecting the efficiency by 5 per cent. New blades can be put in without sending the turbine back to the makers.

Of the two methods of balancing described in Chapter XXVI.,

the Fullagar system has the advantage of reducing the diameter necessary at the high-pressure end of the turbine, and one of the glands has, moreover, to be packed against a very small difference of pressure. On the other hand, the leakage steam from the intermediate dummy passes direct to the exhaust, and with the numbers of packing rings usually adopted, this leakage is fully double that passing the low-pressure dummy. The system therefore seems best adapted for very large units, where the blade lengths being relatively long, the dummy leakage is a small fraction of the total output.

Generally 45 to 50 dummy rings in all are used to wire-draw high-pressure steam down to atmospheric pressure. Hence in the standard design of turbines for generator driving, there are 15 rings on the high-pressure dummy and 15 on the intermediate. The number on the low-pressure dummy is generally about 16. In the case of marine turbines on the three-shaft system, in which the absolute pressure on discharge from the high-pressure turbine is of the order of 40 lb. absolute, the high-pressure dummy may have 27 rings and the low-pressure 20. The dummies are ground in at 80 revolutions, and then slacked back.

Exhaust openings are often smaller than is desirable in view of the high vacua now obtainable, but it is not easy to increase them. The area allowed is often about 1 sq. in. for each 25 lb. of steam passed per hour. Another rule is to use 3 sq. ft. per 1000 horse-power developed.

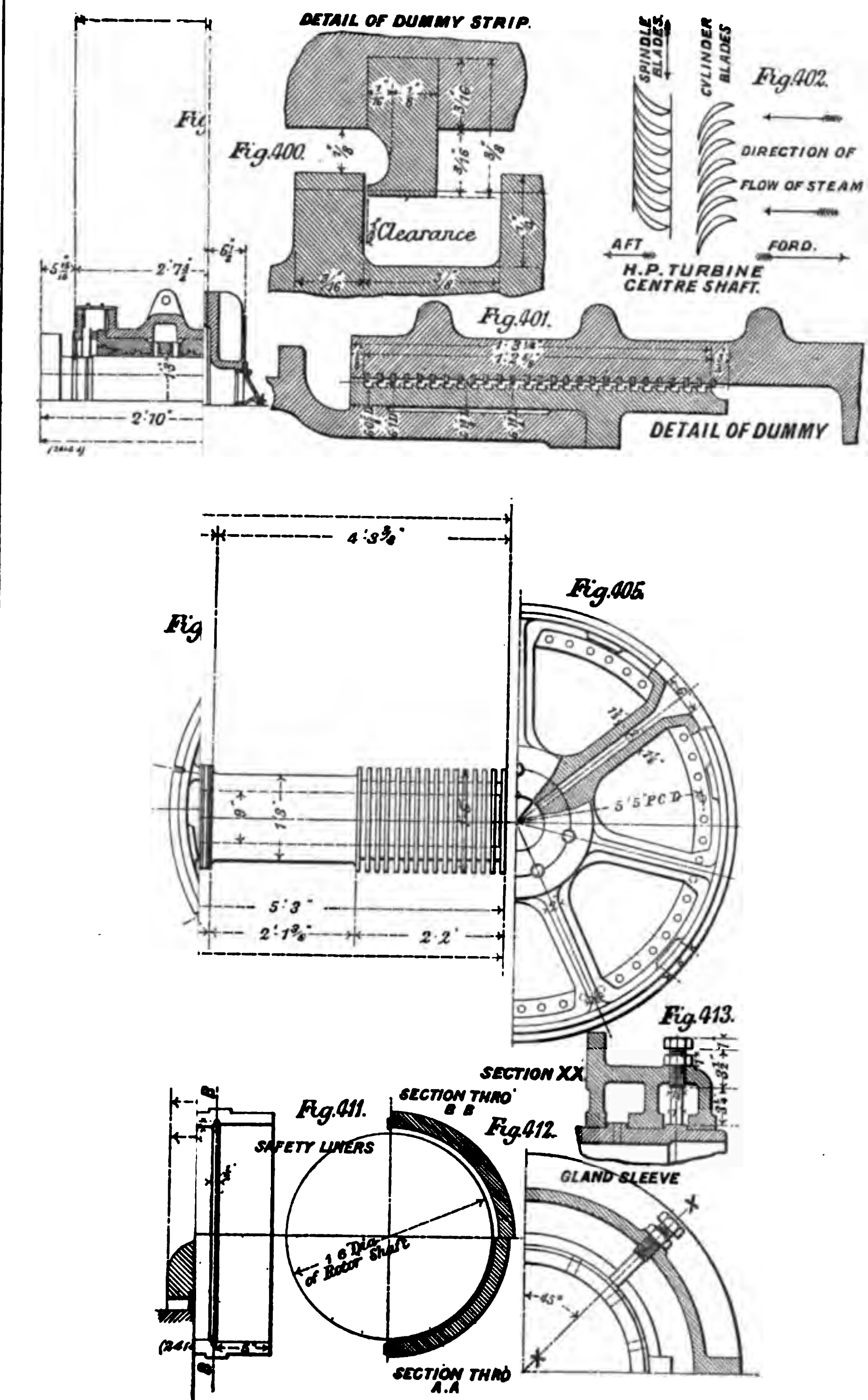
At need, electric-generator turbines of the reaction type can be started up and put on load within one minute, no preliminary warming being required. It is more usual, however, to keep them running light for a quarter of an hour before putting them on load. The main point to be attended to is to get the rotor turning round as soon as possible. The expansion of the casing, so long as it is uniform, involves no danger, and this uniformity is best secured by letting the rotor turn. In the case of marine turbines, where the exhaust is near the top of the casing and not below it, as it is in land practice, this plan is considered inadvisable, and some hours are taken to warm up the turbines before putting them under way.

CHAPTER XXVIII.

THE PARSONS MARINE STEAM TURBINE.

THE adaptation of the steam turbine to ship propulsion involved the solution of a vast number of problems in applied mechanics. The break with previous practice was so abrupt that little assistance was to be obtained from a study of ordinary marine-engine practice. Quite apart from the main difficulty of effecting a reasonable compromise between the conditions necessary for ensuring steam economy in the case of the turbines and those essential to a reasonable efficiency of the screw, there was an infinitude of minor points to be settled connected with the design of the turbine details and with its practical operation. The modern marine engine as we know it embodies the experience of generations of sea-going and manufacturing engineers, the details now considered indispensable to steam economy and ease of handling having been progressively developed. In the course of fifteen years from the date at which the "Turbinia" startled the engineering world by breaking all previous speed records, Parsons marine turbines were built or ordered to an aggregate of some seven million shaft horse-power. The set of marine turbines described in this Chapter were built by the Parsons Marine Steam-Turbine Company, Limited, Wallsend. These turbines were designed to develop 16,500 shaft horse-power when running at 290 revolutions per minute, and to give the boat to which they were fitted a speed of 18 knots. Actually, on trial, some 20,000 shaft horse-power were realised, and in service the boat has proved easily capable of maintaining a speed of 18½ knots with only part of her boilers in use.

A longitudinal section of the high-pressure turbine is represented in Fig. 398, Plate XIX., and a similar section through one of the two low-pressure turbines in Fig. 415, Plate XX. The high-pressure drum (Figs. 403 and 404) is 6 ft. 4 in. in diameter and $1\frac{3}{8}$ in. thick,


DETAIL OF DUMMY STRIP.
Fig.400.
Clearance
SPINDLE BLADES
CYLINDER BLADES
Fig.402.
DIRECTION OF
FLOW OF STEAM
AFT
FORD.
H.P. TURBINE
CENTRE SHAFT.
Fig.401.
DETAIL OF DUMMY
2'.10"
4'.3⅜"
Fig.405.
5'5" P.C.D
5'.3"
2'.2"
Fig.413.
SECTION XX
Fig.411.
SECTION THRO' B B
Fig.412.
SAFETY LINERS
GLAND SLEEVE
of Rotor Shaft
SECTION THRO A.A
45°

and it carries four groups of blades, or "expansions." In each group there are sixteen moving rows of blades, the blade height being $1\frac{5}{8}$ in. in the case of Group No. 1, and increasing approximately in the ratio of 1 to $\sqrt{2}$ from group to group. The blades are fixed by caulking in the usual way. Details of the grooves into which they fit are given in Fig. 399; these are $\frac{5}{16}$ in. deep in the case of the $4\frac{1}{2}$-in. blades, and $\frac{1}{4}$ in. deep for the shorter lengths.

Steam is admitted to the turbine from the belt *a*, and passes through the blading from forward to aft (see Fig. 398). To check leakage in the opposite direction a dummy is provided having a labyrinth packing of twenty-seven fins, as shown in detail in Fig. 401. The fins fixed in the casing fit into grooves turned in the dummy piston, and, as shown in Fig. 400, are turned down to a fine edge where they approach the side of the dummy groove, so that in case of an accidental contact no serious heating can arise. The dummy clearances are adjusted by moving the whole rotor bodily forward or aft, by means of the thrust-block adjusting gear, to be described later on. It will be observed that the dummy is of smaller diameter than the drum, so that there is an unbalanced axial thrust from forward to aft, which is in addition to the thrust on the moving blades themselves. The two together are designed to balance the thrust of the propeller, so that the thrust block has to take merely the difference. The balance between steam thrust and propeller thrust is not equally perfect at all powers, the steam pressure being a little in excess at full power, whilst the propeller thrust preponderates at half-speed The spiders, or wheels, on which the drum is mounted are steel castings, each secured to the drum by three rows of $\frac{3}{4}$-in. screws, there being twenty-one screws per row at the high-pressure end, and twenty-seven per row at the low-pressure end. The screws in the different rows are staggered, and not in line as shown in the sectional drawing. The arms of the high-pressure wheel are hollow, so as to admit high-pressure steam to the hub and to the centre of the hollow shaft. This ensures that the temperature of this shaft and of the corresponding end of the casing shall be approximately the same, and thus have about the same amount of expansion. In this way less fore and aft play is required at the thrust block. The dummy drum is of forged steel, it is turned a sliding fit to its seat on the spider, and is secured to the

latter by fifty-six $1\frac{1}{4}$-in. bolts. The flange is tapped in four places to take starting screws, should it ever be desirable to break the joint.

The spider at the low-pressure end has solid arms, but, as will be seen from Fig. 403, it is slotted at nine points round its circumference. These slots serve to prevent any lodgment of moisture on the interior face of the drum, allowing any condensation there to be drained off. Details of the high-pressure casing are illustrated in Figs. 424 to 431, pages 318 and 319. It consists of two principal castings, forming respectively the upper and lower halves of the turbine cylinder. There is, it will be seen, an auxiliary steam belt *c*, Fig. 424, through which steam can be by-passed

Fig. 420

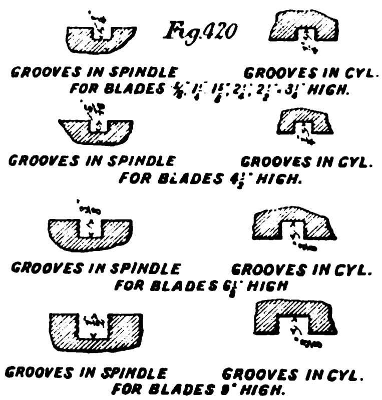
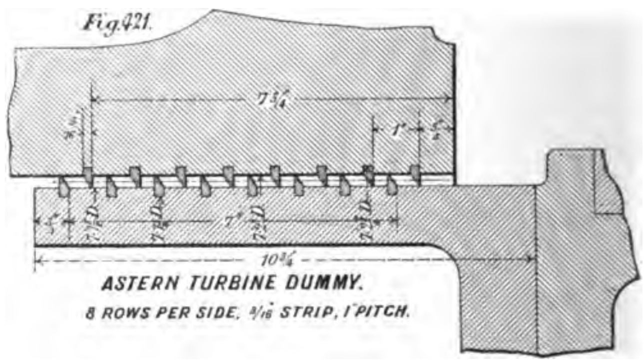


Fig. 421

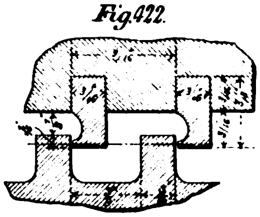


to the beginning of the second group of the turbine. A couple of pipes, one of which connects the two seatings, shown dotted at *d* and *e*, Fig. 424, supply this steam belt. The correct position of these by-pass seatings is shown at *e*, Fig. 425 (see also Fig. 426), and there is a similar set of seatings on the right-hand side of the upper cylinder casting. There are also two main steam inlets situated near the bottom of the casing, as indicated at *f*, and similarly there are two exhaust branches at the after end of the cylinder. One of these is shown dotted in Fig. 424, and in its correct position on the right-hand side of Fig. 426.

The drum which carries the dummy rings is, it will be seen, separate from the main castings, being bolted on a turned seating, as indicated in Fig. 426. A half section through this dummy drum is reproduced in Fig. 427. A brass strip $\frac{3}{16}$ in. thick covers the

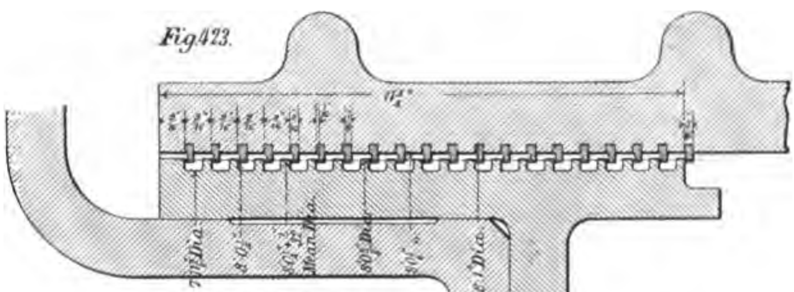
horizontal joint, as is indicated at *g*. Heavy brackets, to transfer the weight of the turbine to the pedestals in the ship, are cast solid with the main castings, as best seen to the right of Fig. 425. Manholes at each end enable the interior of the turbine to be examined without the necessity of raising the cover. The glands by which the shaft is packed are mounted in separate castings, which are secured to turned seats at each end of the cylinder. Details of these glands are illustrated in Figs. 406 to 413, Plate XIX. As shown in Fig. 406, the labyrinth is of the radial-fin type, these fins being shown to an enlarged scale in Fig. 407. At the outer end of the gland there are five Ramsbottom rings, which prevent leakage of the steam into the engine room. The bushing, which carries the stationary fins, is of gun-metal, and is made a good fit to the outer

Fig. 422.



DETAIL OF DUMMY STRIP.

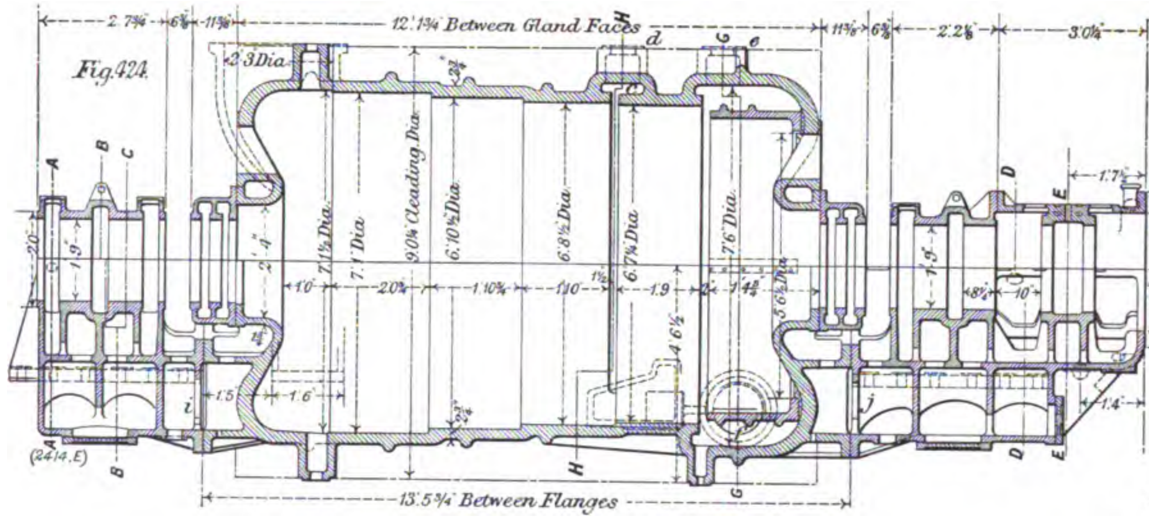
Fig. 423.

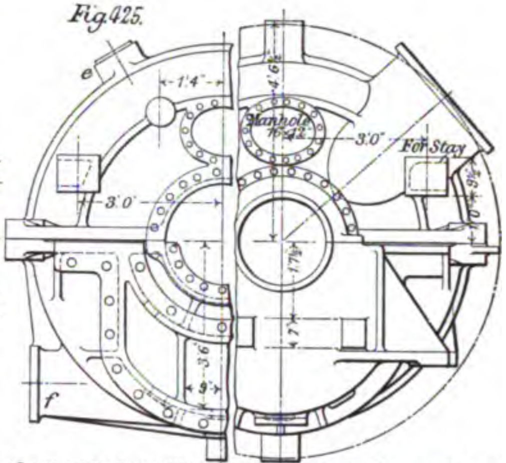


L P AHEAD DUMMY
20 ROWS, 3/16 STRIP, 3/16 PITCH.

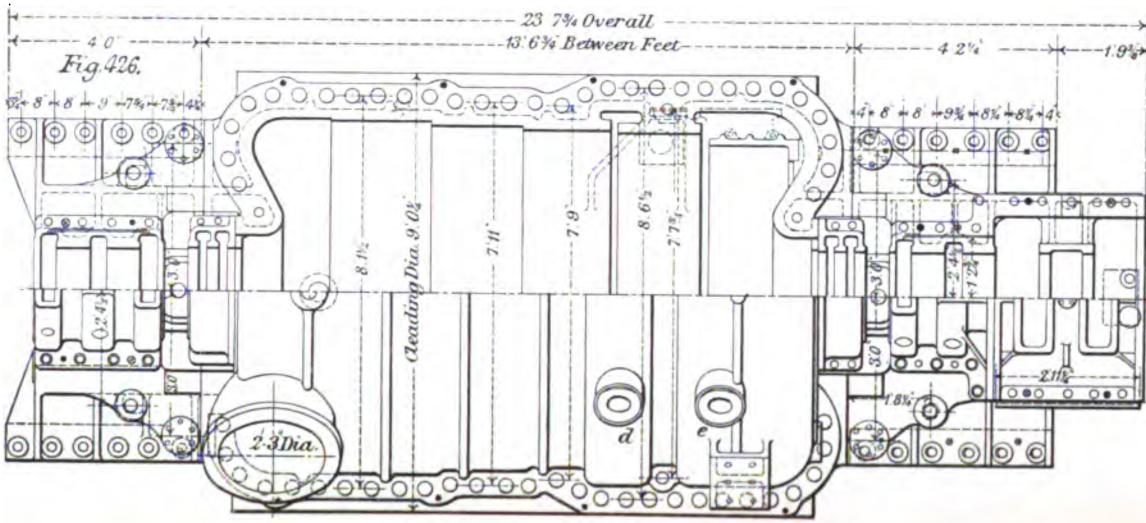
cast-iron sleeve, in which there are two steam or leak-off pockets, as indicated. These pockets are put in communication with the interior of the packing by fourteen $\frac{3}{4}$-in. holes drilled through the sleeve, as shown in Fig. 406. The uppermost of these holes are tapped for lifting purposes. The sleeve is shown separately in Figs. 410 and 411. Safety rings are fitted at A, A and B, B (Fig. 410) to prevent the fins fouling when removing the lower half of the gland. Starting screws are provided as in Figs. 412 and 413, to free the sleeve from the outer cast-iron holder, when it is desired to open the gland.

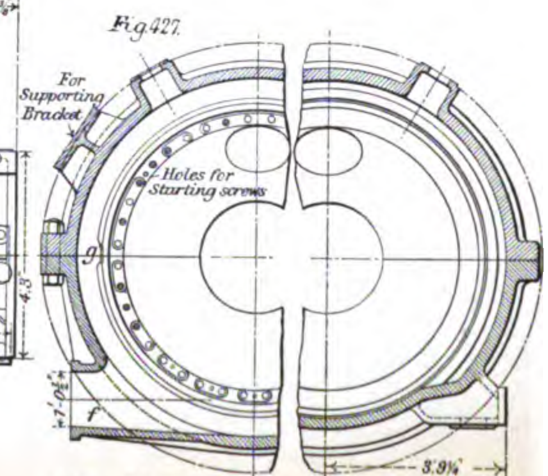
The aft and forward bearings are supported by strong brackets bolted to the end covers, as shown in Fig. 424. At the forward end there is a thrust block as well as the ordinary bearing. A view of this forward bearing to a somewhat larger scale is reproduced in Figs. 446 and 447, page 324. The oil is fed into the bearing at the





FORD END OF CYLR LOOKING AFT. AFT END OF CYLR LOOKING FORD





SECTION G.G. LOOKING FORD SECTION H.H. LOOKING FORD

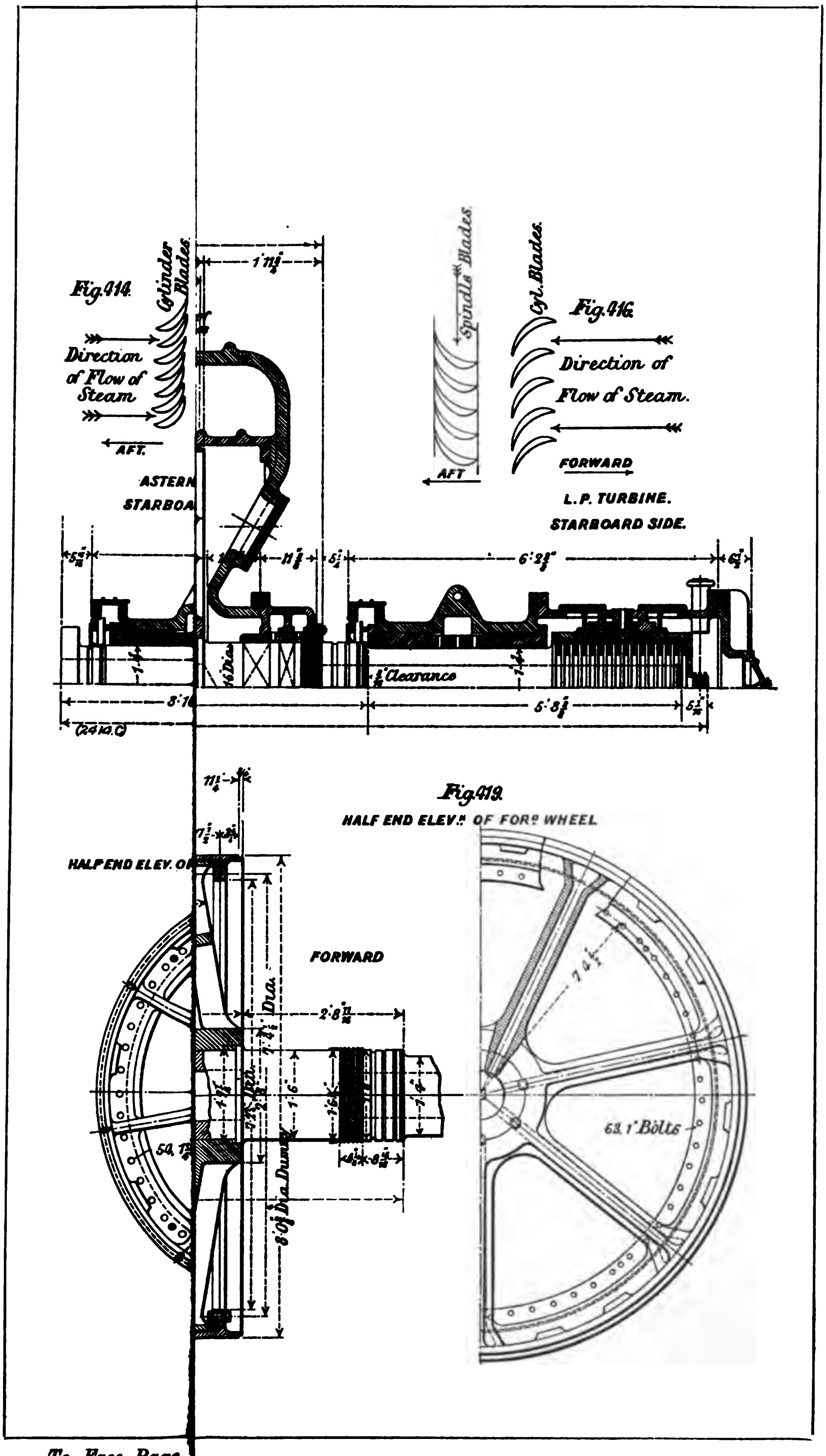
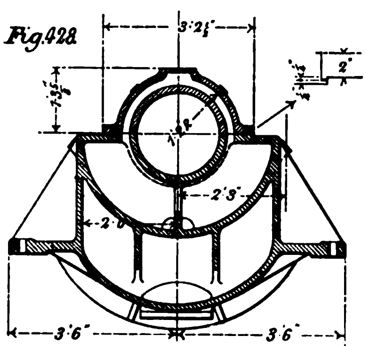


Fig. 414.
Cylinder Blades.
Direction of Flow of Steam
AFT.
ASTERN
STARBOA
Spindle Blades.
Cyl. Blades.
Fig. 416.
Direction of Flow of Steam.
AFT
FORWARD
L. P. TURBINE.
STARBOARD SIDE.
Clearance
Fig. 419.
HALF END ELEV.N OF FORD WHEEL
HALF END ELEV. OF
FORWARD
63, 1" Bolts
Dia.
Dia. Dum

To Face Page

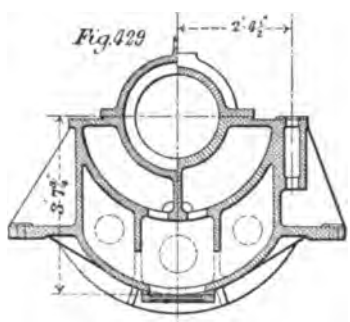
centre, and escapes at both ends, where it drains down into the bottom of the bracket. A wiper, shown at *h*, Fig. 446, prevents any creeping of the oil along the shaft, and, as a further safeguard, a few serrations are also turned in the latter, as indicated, to an enlarged scale in Fig. 451. The bushes are lined with white metal, but large safety strips are provided at each end to prevent injury should the white metal be melted out through a failure in the lubrication. The oil is caught in wells formed in the bottom of the supporting brackets, and

Fig. 428



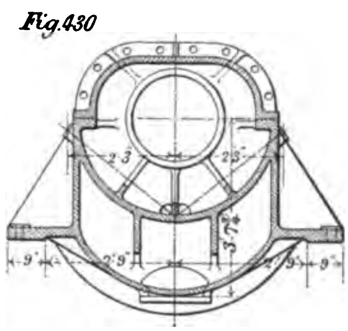
SECTION A.A. LOOKING FOR^{D}

Fig. 429



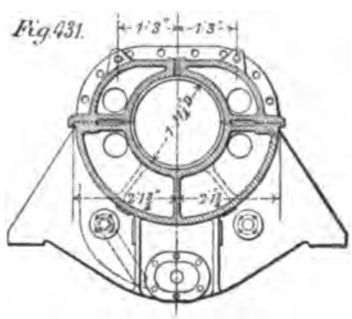
SECTION B.B. LOOK^{G} FOR^{D} SECTION B C. LOOK^{G} FOR^{D}

Fig. 430



SECTION D.D. LOOKING AFT.

Fig. 431



SECTION E.E. LOOKING AFT.

is drawn off, filtered and again passed through the bearings. Any escape of water from the gland is caught in separate pockets, shown at *i* and *j*, Fig. 424.

The low-pressure rotor, Figs. 415 and 418, Plate XX., is in essentials very similar in its construction to the high-pressure turbine rotor. It is, however, as shown, provided with a stiffening ring near the centre of its length, which is not required in the case of the smaller high-pressure drum. Grooves formed in the circumference of this ring prevent water-collecting on the forward side of it. The rotor has

eight groups of blades or "expansions," the blade height at the inlet end being $1\frac{5}{8}$ in., and at the exhaust end 9 in. As usual in the case of low-pressure marine turbines, the last three groups have the same blade height, the necessary increase in the steam way being obtained by the use of "semi-wing" and "wing" blades, the opening through which is much greater than is the case with normal blades. The low-pressure dummy is much smaller than the main drum, since, although each screw develops the same thrust, the axial pressure of the steam in the blading is much less in the low-pressure than in the high-pressure cylinder, and this requires to be supplemented by necking down the dummy to a greater extent. As before, it will be seen that provision against the possibility of unequal expansion of drum and casing is made by leading steam through the hollow arms of the forward spider to the centre of the shaft. The packing for the low-pressure dummy is shown in Figs. 422 and 423, page 317, but, since the steam has a large specific volume, there are only twenty rows of fins instead of twenty-seven.

As will be seen from Figs. 415 and 418, the low-pressure drum is extended to form the drum for the reverse turbine, the casing for which is bolted on to the main turbine casing, as shown in Fig. 415. This reverse turbine is bladed with four groups of blades, each consisting of nine rows, and varying in length from $\frac{5}{8}$ in. up to $2\frac{1}{2}$ in. in height. The whole of the expansion of the steam is thus effected with but thirty-six rows of moving blades, whilst in the main turbines there are sixty-four moving rows in the high-pressure turbines and an equal number on the low-pressure drum. Hence the reverse turbine is much less efficient in its use of the steam; but this is, of course, a minor consideration, as a ship is never called upon to steam for a long time with her engines reversed. For the astern turbine dummy a radial fin packing is used, and the arrangement of this is shown in detail in Fig. 421, page 316. This form of packing permits of free differential expansion between the casing and the rotor of the turbine, but somewhat larger clearances are necessary than with the form of packing used for the ahead dummies, and these clearances are, moreover, not adjustable.

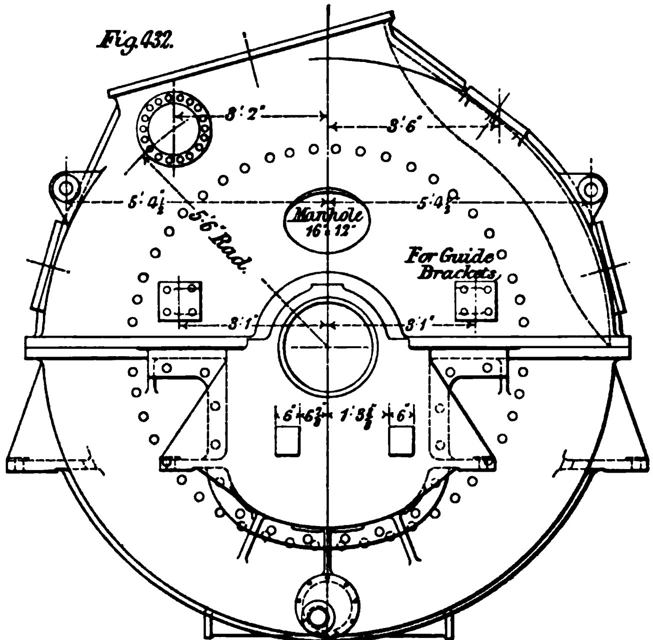
Detailed views of the low-pressure casing are given in Figs. 432 to 434, page 322, and in Figs. 435 to 443, Plate XXI. The main casing, it will be seen, is made in four parts, there being a

circumferential joint near the centre of the length. A view of this joint is given in Fig. 444, page 323, and on the right is a detail showing how the studs are arranged in the neighbourhood of the main horizontal flange. The main steam supply enters through a branch on the upper half of the cover, but for manœuvring purposes an auxiliary supply can be passed in through a branch cast with the lower half of the casing, as indicated in Figs. 435 and 438. The exhaust branch, 5½ ft. by 6½ ft., is supported against collapse by forged stays, shown in position in Fig. 439. The casing for the astern turbine is bolted inside the main casing. Its forward end rests on a machined support, as shown in Figs. 433 and 435, and its steam supply is derived from a branch which passes through one of the walls of the main exhaust port, as best seen in Fig. 433. Weep holes to keep this astern cylinder free from condensation are provided in the steam belt, as shown in Fig. 435.

The brackets which carry the bearings and thrust block are very similar to those used for the high-pressure turbine, and cross sections will be found in Fig. 434 and in Figs. 440 to 443, Plate XXI. The bored guide shown at the top in Fig. 434, and also in Fig. 435, takes the governor spindle, which is driven by helical gearing; the driving gear being keyed to the end of the main shaft, as shown in Fig. 415, Plate XX.

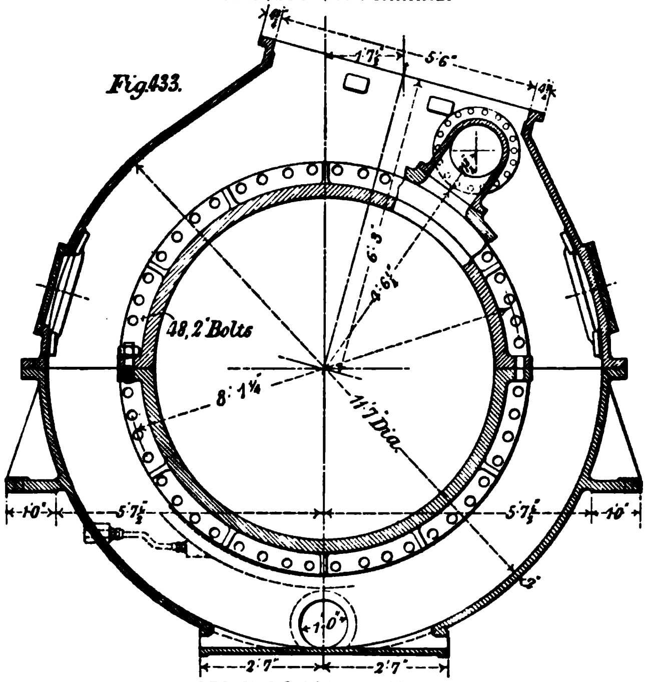
Some further details of the thrust block and its adjusting gear are given in Figs. 453 to 468, pages 326 and 327. The upper and lower halves of the block are independently adjustable. The two are shown separately in Figs. 455 and 456, page 326. Each half is provided with strong lugs, taking the bolts shown in Figs. 453, 454, and 457, and also separately in Fig. 465, page 327. These bolts are screwed at the one end, where they pass through a bronze nut which turns in a bored seat formed on the front cover of the thrust block casing. Each of these nuts at its outer end is formed into a worm-wheel, as best seen in Fig. 466; and the upper pair can be rotated simultaneously by a worm (see Figs. 453 and 454), so as to move in or out the top half of the thrust block. A similar provision is made for setting forward or aft the lower half of the thrust block. In this way the rotor can be moved to and fro inside its casing, and the dummy clearances adjusted as desired. When this has been done, permanent liners are inserted, as shown at

Fig. 432.



END VIEW. LOOKING FORWARD.

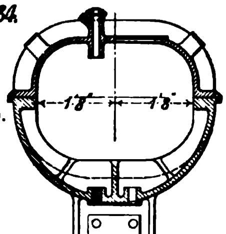
Fig. 433.



SEC. AT C.C. LOOKING AFT

Fig. 434.

HALF SEC. THRO' H.H. LOOKING FORWARD.

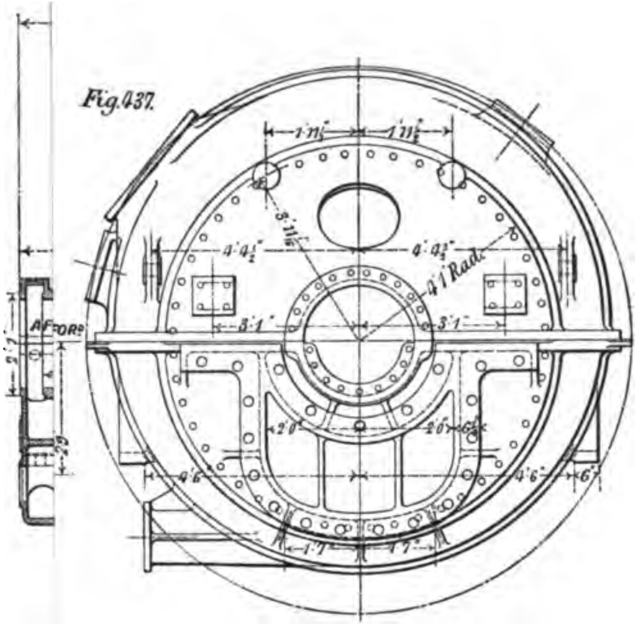


HALF SECN THRO' I.H.

PLAN OF FACE

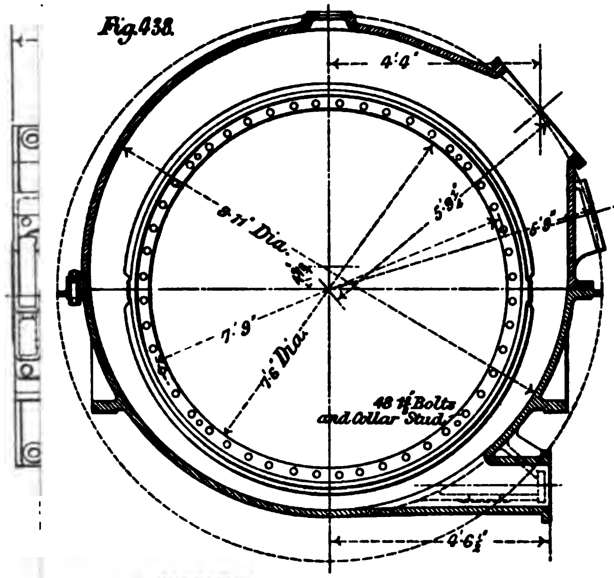
E.

Fig. 437.

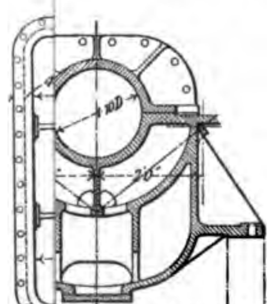


FORWARD END OF CYLINDER
LOOKING AFT.

Fig. 438.

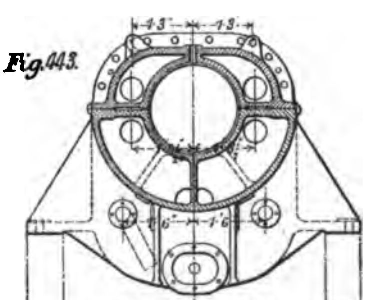


SECTION AT E.E. LOOKING FORWARD.



...RO' F.F. LOOKING FOR?

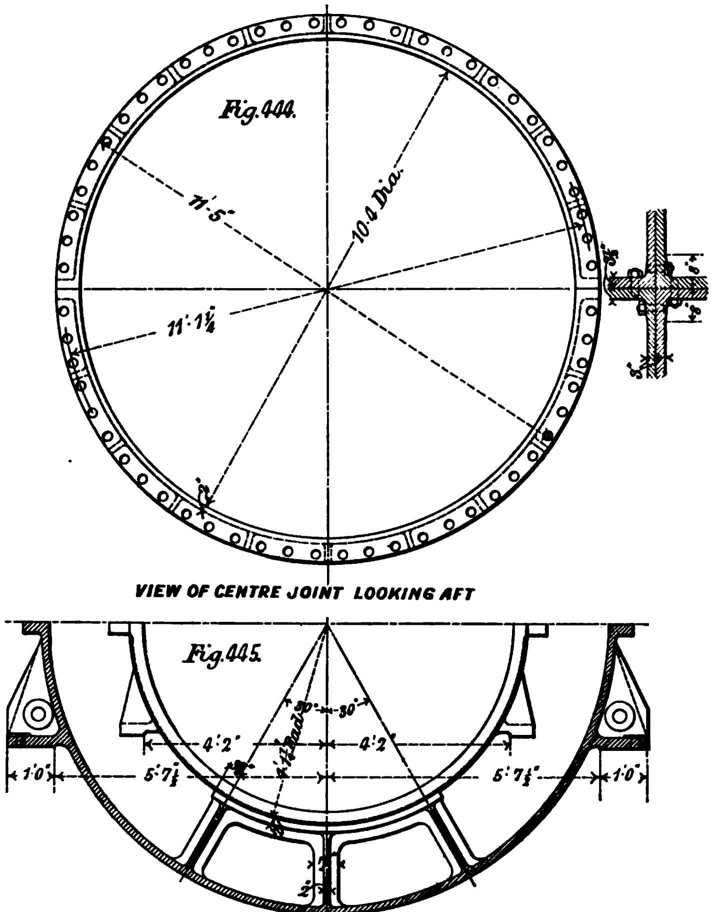
Fig. 443.



SECTION THRO' G.G. LOOKING AFT.

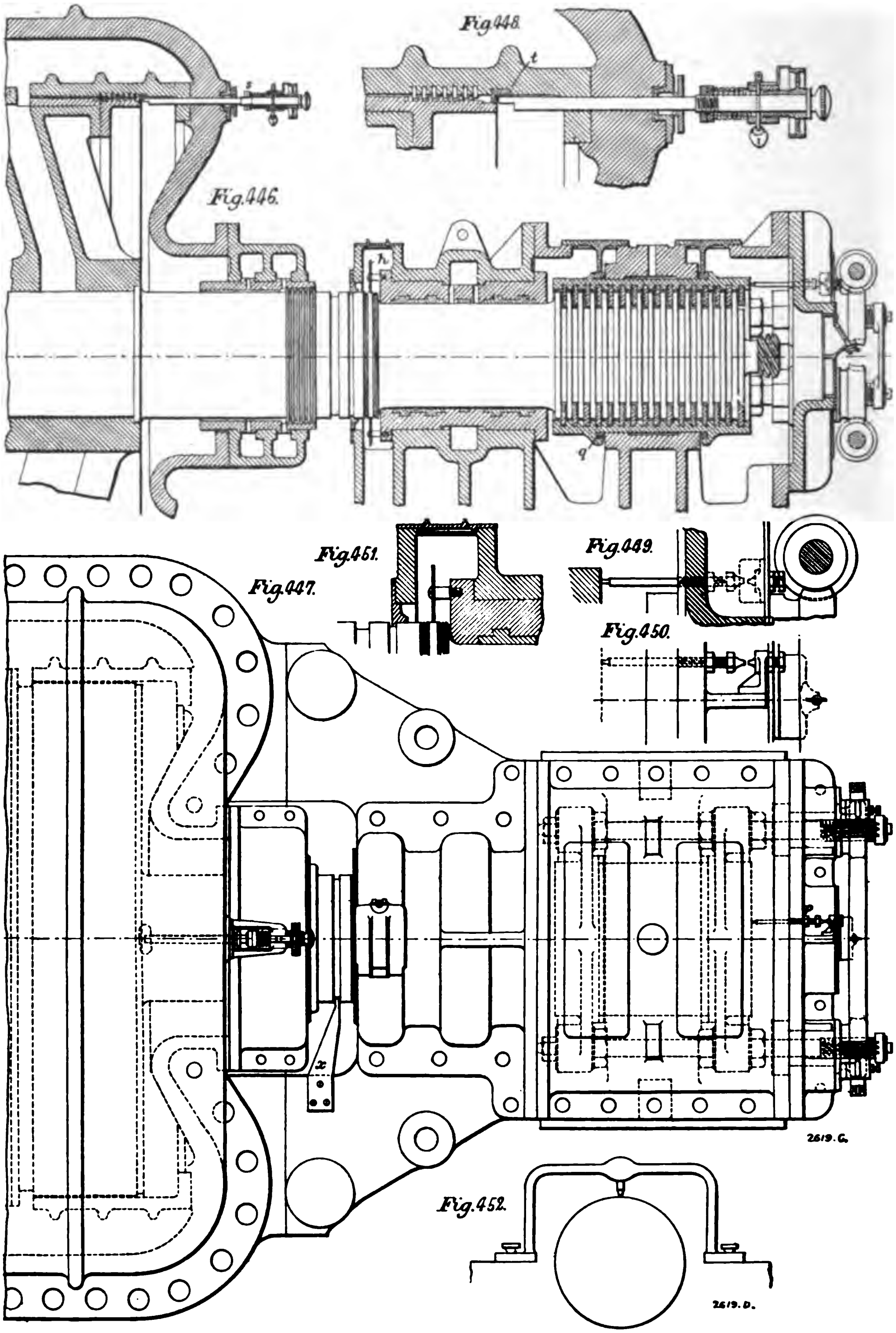
To Fa

w, *w*, Fig. 457, and the pair of screws locked by putting over their squared ends the locking bar shown separately in Fig. 459. Any thrust from aft forward is taken on the lower half of the thrust block, whilst the upper half takes any pull in the opposite direction.



Figs. 444 and 445. Sections through Casing.

To gauge the dummy clearances two indicators are provided. Of these, one is a finger piece, fitting into a groove turned in the shaft, as shown at *x*, Fig. 447, page 324. This is fixed after the clearances have been finally adjusted, and the clearance between it and the side of its groove, as determined by feeler gauges, is recorded. This being always accessible, clearances can be readily and easily checked subsequently, but owing to differences in steam temperatures the clearances at this finger-piece and at the dummy are not always



Figs. 446 to 452. Micrometer Finger and Bridge Gauges.

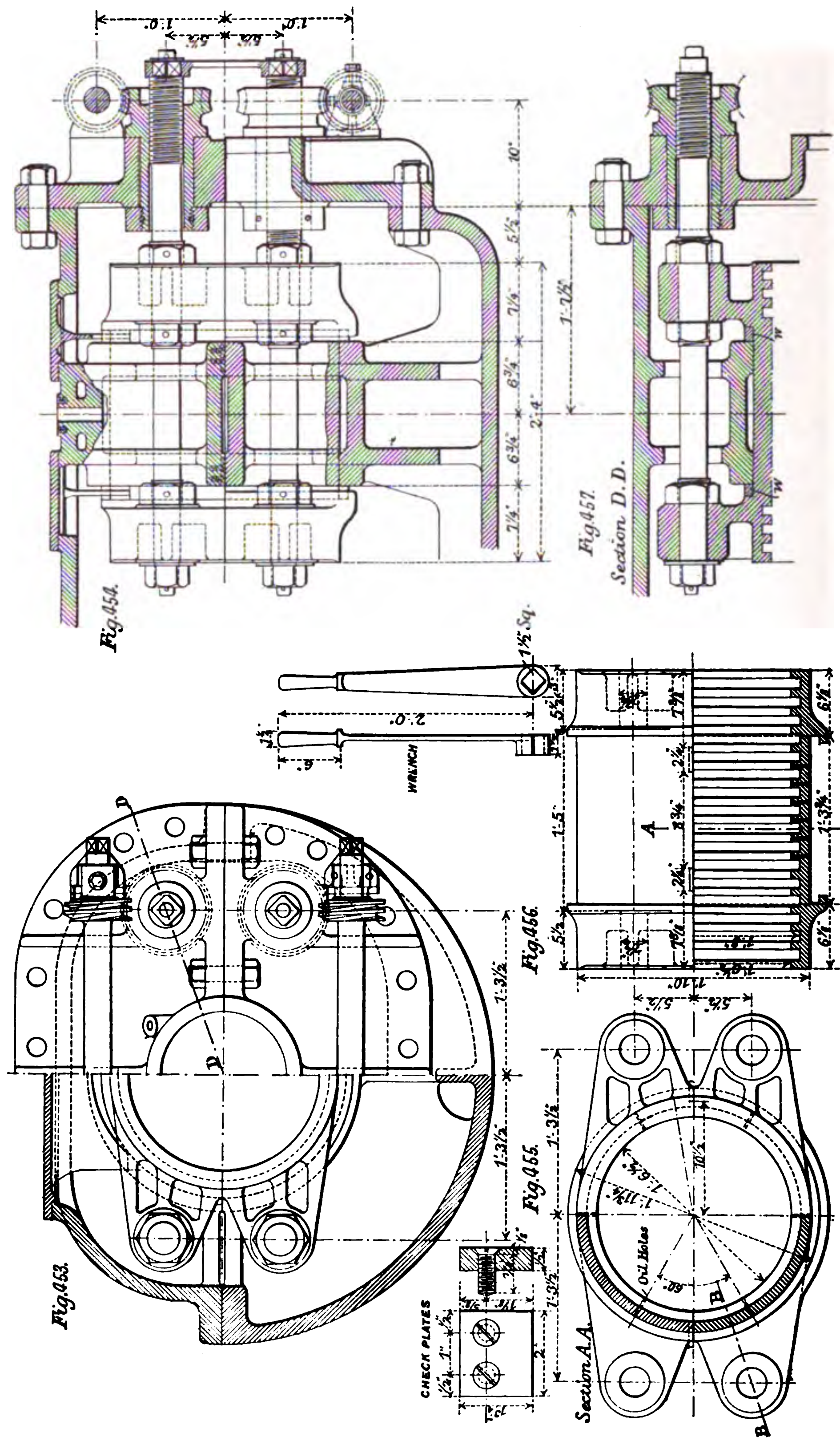
in agreement with each other. A second means of checking the dummy clearance is therefore provided. This consists of a micrometer gauge, fitted as shown at *s* (Fig. 446, page 324), and to a larger scale in Fig. 448.

Normally the stem of the micrometer is locked by a cotter, as indicated; in which case it will be seen its inner end stands perfectly clear of the adjacent stop *t*, Fig. 448, which serves as reference datum. In using the instrument the cotter is withdrawn, and the inner end of the stem allowed to rest against the fixed stop *t*. The micrometer wheel is then turned till it just moves the stem again, and a reading taken. By means of the knob at its outer end the stem is now turned through two right angles, in which it will be seen its extreme end clears the datum stop *t*, and it can then be moved inwards until it comes into contact with the rotating dummy-ring. The micrometer wheel is again turned so as to just bring the stem clear, and a second reading taken. The difference between these readings gives the dummy clearance in mils. A final check reading on the stop is generally taken to provide for the possibility that the bottom of the stem may have been worn away a little during its contact with the moving dummy.

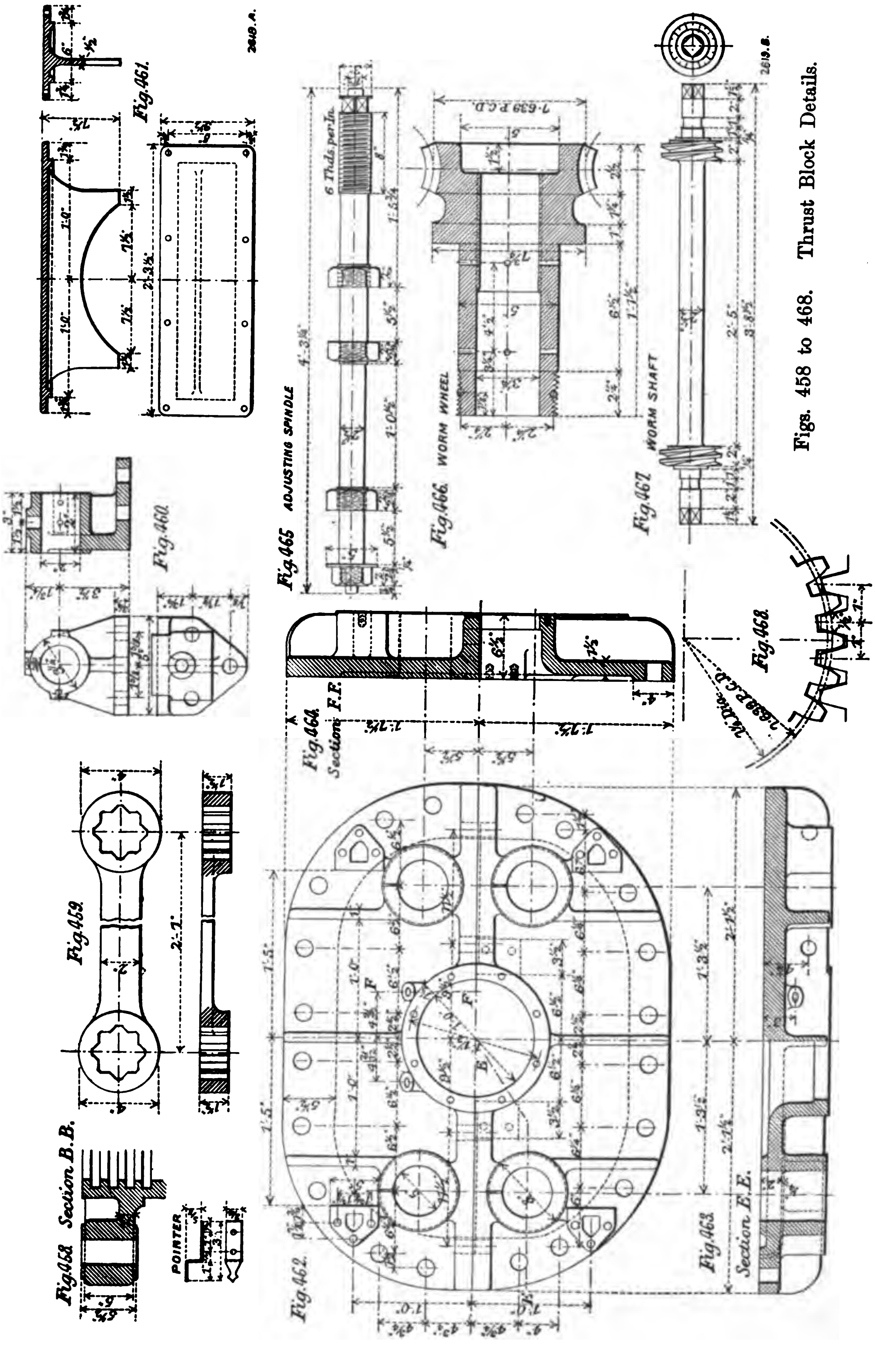
A third gauging point is provided at the thrust block, as indicated at *p* in Fig. 447, and shown to a larger scale in Figs. 449 and 450. This is used to adjust the "oil" clearances between the collars of the thrust block. After the lower thrust block has been adjusted to give the desired dummy clearance, and brought up hard against its aft liner *q*, Fig. 446, the upper block is in turn moved forward until the collars jam. The clearance between the gauge points in Figs. 449 and 450 is then determined by feeler gauges, and the top half is then slacked back 12 mils, so as to give space for the oil to get between the bearing surfaces.

To test the bearings for wear, the "bridge" gauge shown in Fig. 452 is employed. This fits on to machined seats, and straddles the shaft as shown. Feelers placed between the hardened gauge point and the shaft enable any wear on the bearings to be readily detected.

The general arrangement of the turbines, together with the pipe system and manœuvring gear, is represented in Figs. 469 to 472, pages 328 to 330. The turbines are arranged on the



Figs. 453 to 457. Thrust Block for Marine Steam Turbine.



Figs. 458 to 468. Thrust Block Details.

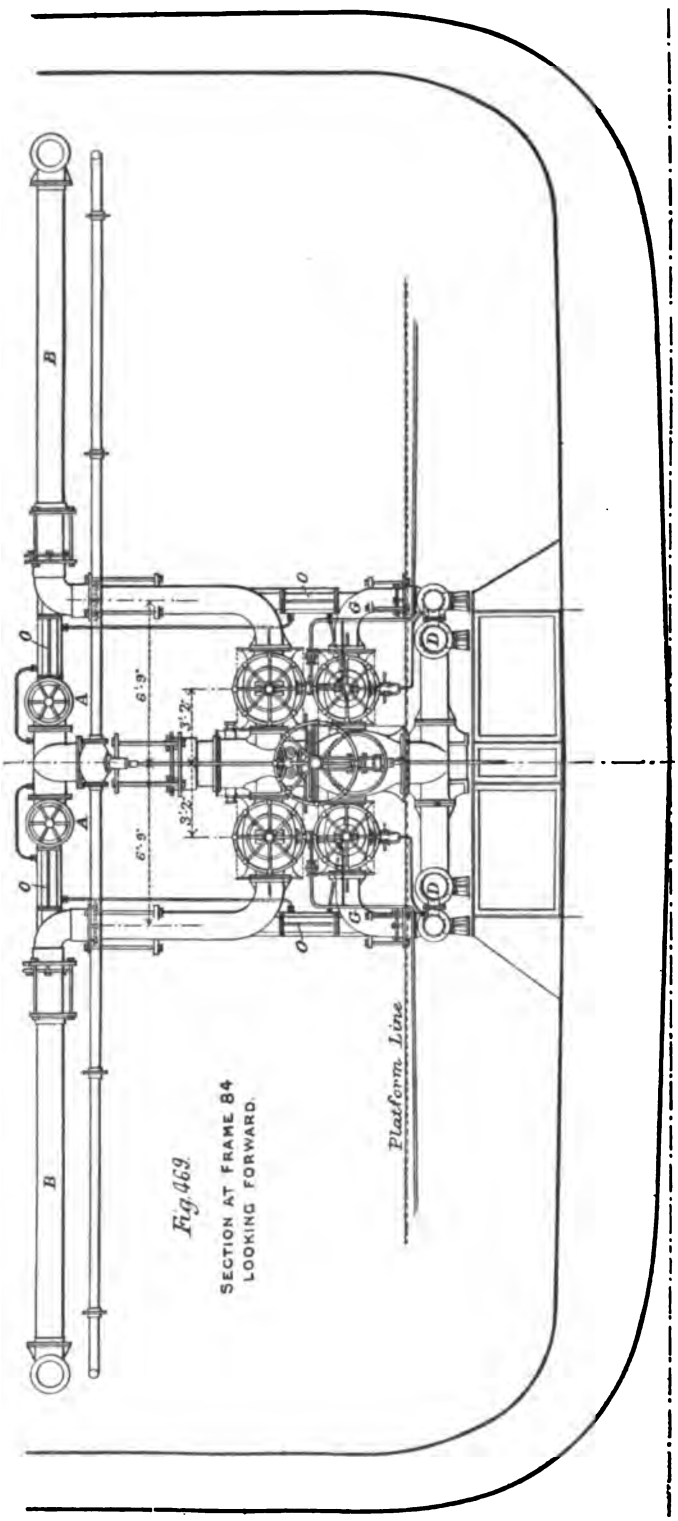


Fig. 469. Manœuvring Valves for Marine Steam Turbine.

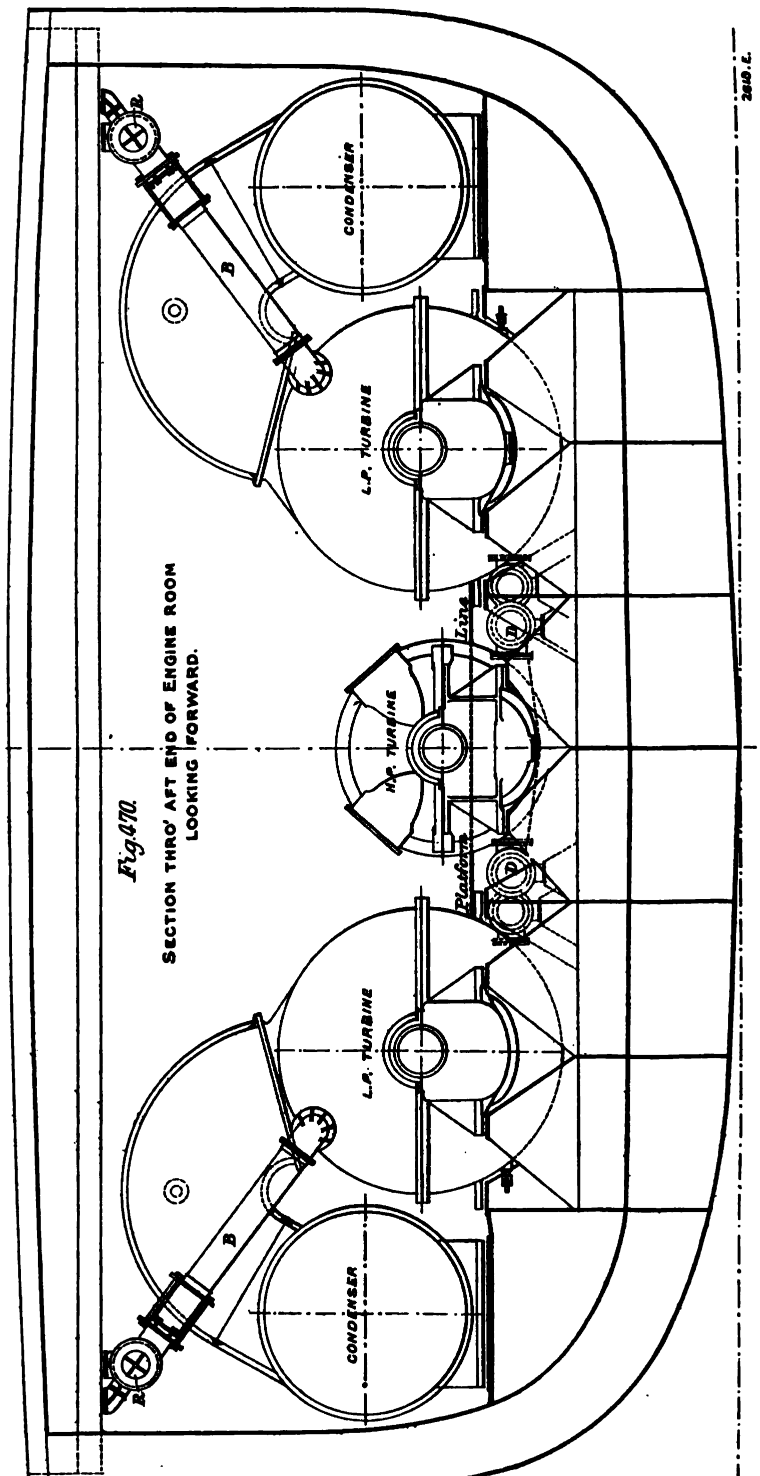


Fig. 470. Ship Arrangement of Marine Steam Turbines.

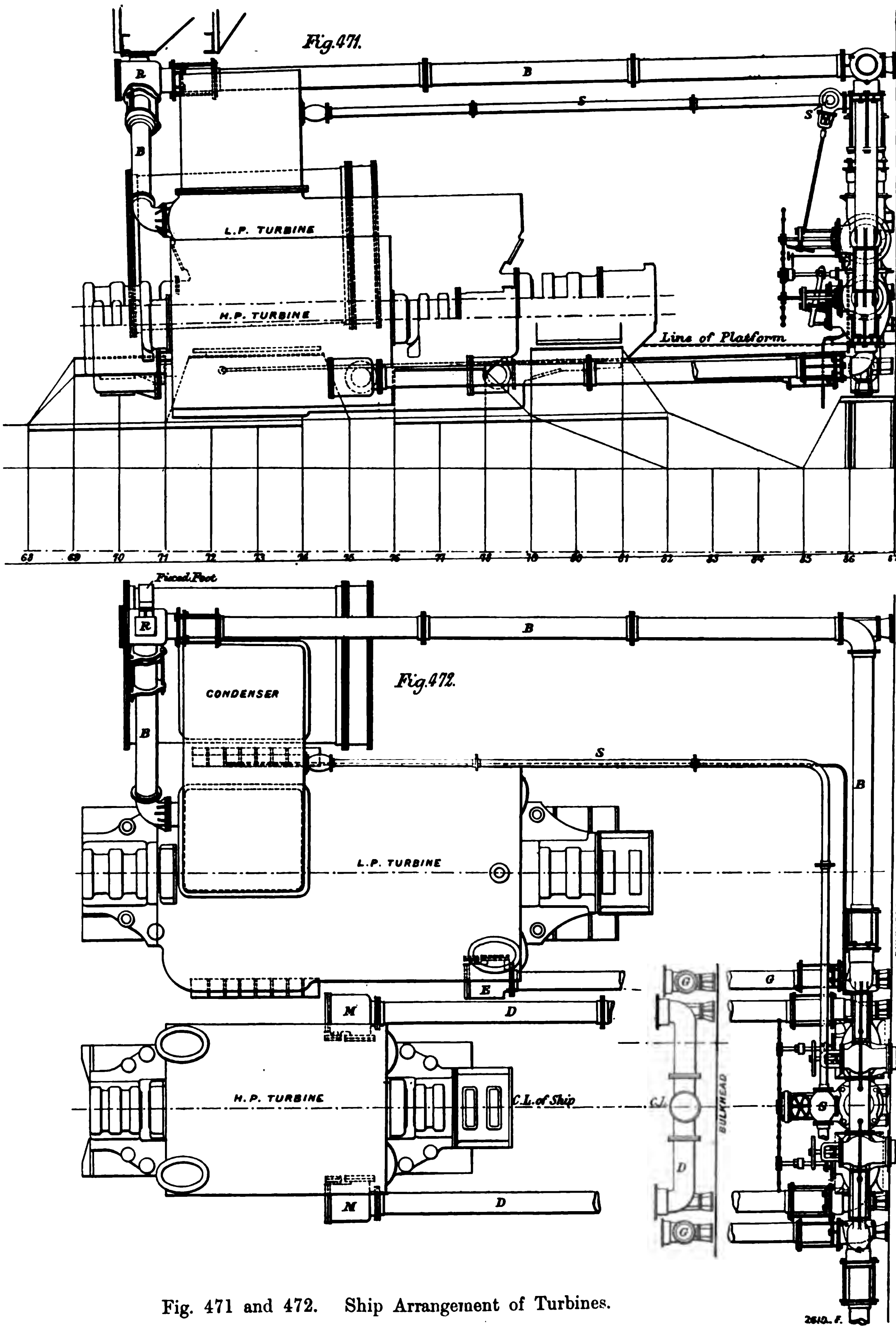


Fig. 471 and 472. Ship Arrangement of Turbines.

three-shaft system, a high-pressure turbine, driving a central shaft, exhausting into two low-pressures placed in the wings, and through which the steam passes in parallel. As usual at sea, the condensers are at a higher level than the turbines, and the exhaust pipe has accordingly to be fitted above instead of below the turbines.

The steam from the boilers enters the engine-room through two bulkhead stop valves A, A, Fig. 469, it then passes through the T-piece shown, and an expansion sleeve, into an 18-in. main regulator valve, the hand wheel of which is the central one of the group of five, shown in Fig. 469. On each side of this central valve are two manœuvring valves. These are used when working a boat in or out of harbour, at which time the main regulating valve is closed. The upper, when open, admits steam to the reverse turbine on the same side of the ship through the pipes B, B.

The lower valves, on the other hand, let, through the pipes G G, high-pressure steam direct into the main low-pressure turbine on the corresponding side of the ship, so that, if desired, one of the wing screws may be run ahead, whilst the other runs astern. When the boat is clear, the manœuvring valves are closed and the main regulator opened, and the steam then enters the high-pressure turbine through the pipes D, D. The pipe D is supported at its ends both from the bulkhead (see Fig. 469) and from the platform plating; but the bolts which secure it are in enlarged holes, and are not tightened down sufficiently to prevent free expansion. Strainers E and M, Figs. 472, are provided where the main steam pipes are led into the turbine casings. The strainer for the astern turbine steam is at R. The valve and pipe system S leading to the condensers is used for silently blowing down the boilers. The struts shown at O, Fig. 469, are hollow, and the interior is in each case connected up to a supply of steam, so as to keep it at the same temperature as the pipes proper, and thus minimise expansion strains. Drain pipes $\frac{3}{8}$ in. in diameter are provided to keep clear of water all the large valves where there is any possibility of moisture collecting, and for warming up the main turbines a small pipe system, 1 in. in diameter, and fitted with the necessary valves, is also provided, the steam being taken from above the main regulator valve.

CHAPTER XXIX.

DISC-AND-DRUM TURBINES.

A FORM of turbine known as the disc-and-drum type was originated about 1905 by the British Westinghouse Company, and has since been largely adopted. This type of turbine consists of one or more velocity-compounded wheels, followed by a drum carrying either impulse or reaction blading. When this type of turbine is used for generator driving, it is usual for the velocity-compounded wheel to have only two rows of moving blades, but when this variety of turbine is to be coupled direct to a screw propeller the wheels carry three or even four rows of moving blades, though this necessarily involves a substantial sacrifice of turbine efficiency.

The advantages of the disc-and-drum machine are practically wholly mechanical in character. It is not possible, at least in the case of large units, to obtain with them as high an efficiency as is practicable with turbines constructed on the reaction principle throughout. The velocity-compounded wheel, however, replaces the whole of the high-pressure end of the latter, which, being necessarily of small diameter, requires many rows of blades, involving at times a lack of stiffness in the motor. Whilst much shorter than a machine constructed wholly on the reaction principle, the internal construction is practically as simple, when but one velocity-compounded stage is used. No diaphragms are required, and the rotor is simple in form and very stiff, and is as readily accessible as in the case of an ordinary reaction machine. An additional advantage is that since about one-third of the total output is produced by the velocity-compounded stage, the steam temperature is considerably reduced before it enters the reaction section, extremes of pressure and temperature being confined to a mere nozzle box. Since, however, there is, as yet, no adequate theory

of the design of velocity-compounded wheels, reliance has to be placed wholly on empirical data, and failures to realise guarantees have been very frequent with disc-and-drum machines.

A large disc-and-drum machine, constructed by Mr. Franco Tosi, of Legnano, Italy, is illustrated in Figs. 473 and 475, pages 334 and 336. The machine was designed to develop 4500 horse-power when running at 1500 revolutions per minute, but "on overload" it can generate 5600 horse-power for an indefinite time. The casing is of cast iron, and the rotor shafting of forged steel. Owing to the adoption of a velocity-compounded stage, the total distance between bearings is, as explained above, very short. The turbine has been designed to work with high steam pressures and high superheats, and hence in constructing the rotor "shrunk" fits have been avoided and the drum secured to the stub ends by flanged joints. These, it will be observed, are registered against both external and internal surfaces, so that in case the drum heats quicker than the flange, it is still tight on the latter, and cannot get out of truth.

The governing gear forms a special feature of the turbine. The steam supply is led to a nozzle box from a Ferranti-type stop valve. In the nozzle box are four independent valves, each of which controls the supply of steam to a separate group of nozzles. Of these, three are used at full load, whilst the fourth comes into action when an overload has to be taken. One of these valves can be seen in position at the top left-hand corner of Fig. 473, and their arrangement on the end of the casing is also illustrated by Fig. 474 and the key view, Fig. 479, page 336. They are of the equilibrated double-beat type, and their construction is illustrated in detail by Figs. 475 to 478, page 336. They are operated by oil relays controlled by the governor. Details of this gear are shown in Figs. 480 to 484, page 337. The governor, Fig. 480, operates a piston valve d, which controls the passage of oil from the port a to the port b. In the position shown the communication between the two is fully open, and the oil leaving at b would pass to the valve boxes, and entering the latter by the port R, Fig. 475, page 336, would lift the valve against the pressure of the spring shown above the oil piston. The valve, it will be seen, is of the double-beat type, and, as shown in Fig. 475, is made in two parts, so that it can be

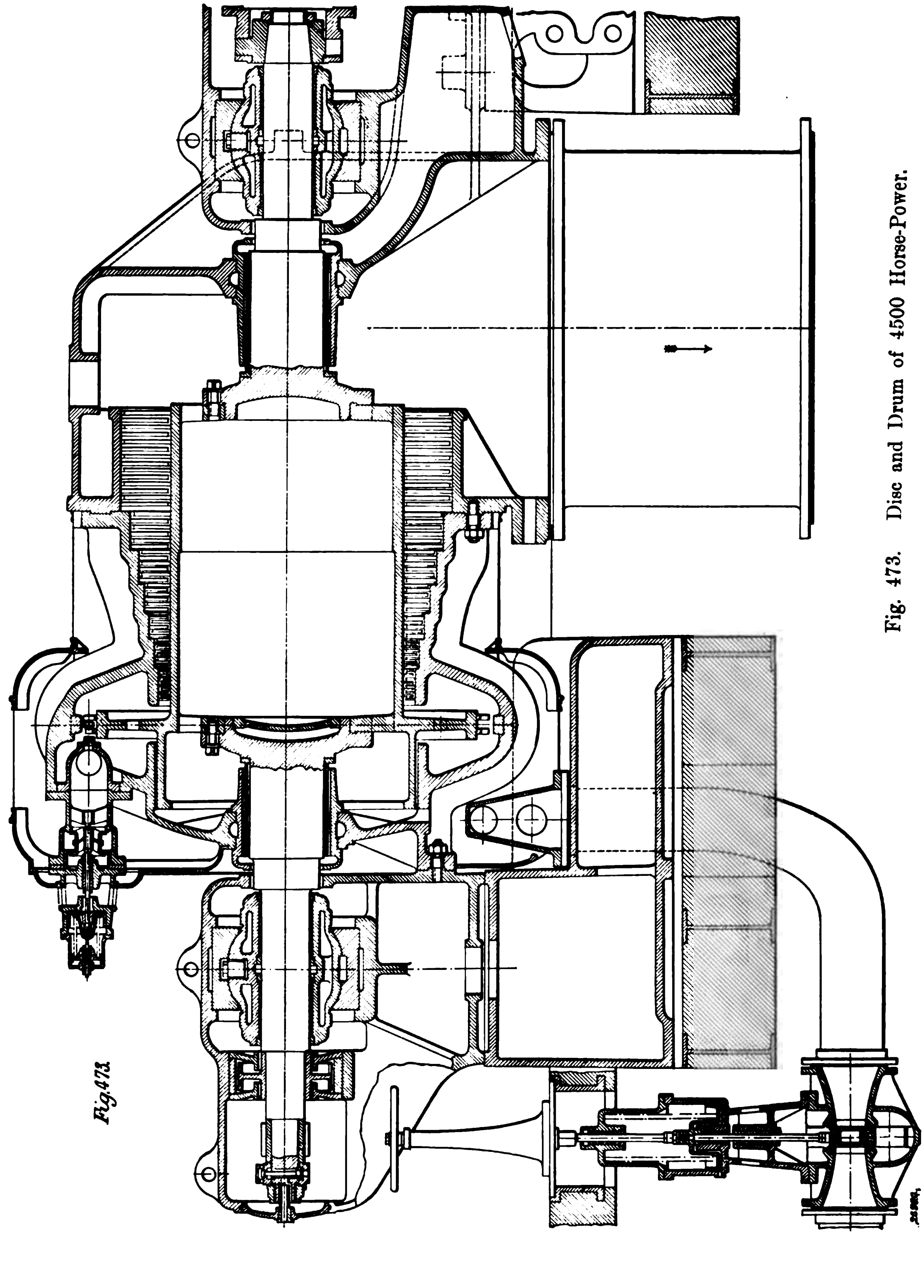


Fig. 473. Disc and Drum of 4500 Horse-Power.

fully equilibrated. Oil leaking past the oil piston escapes by the port T back to the pump. If it leaks off faster than it is supplied through the port R, the valve is closed by the action of the

Fig. 474.

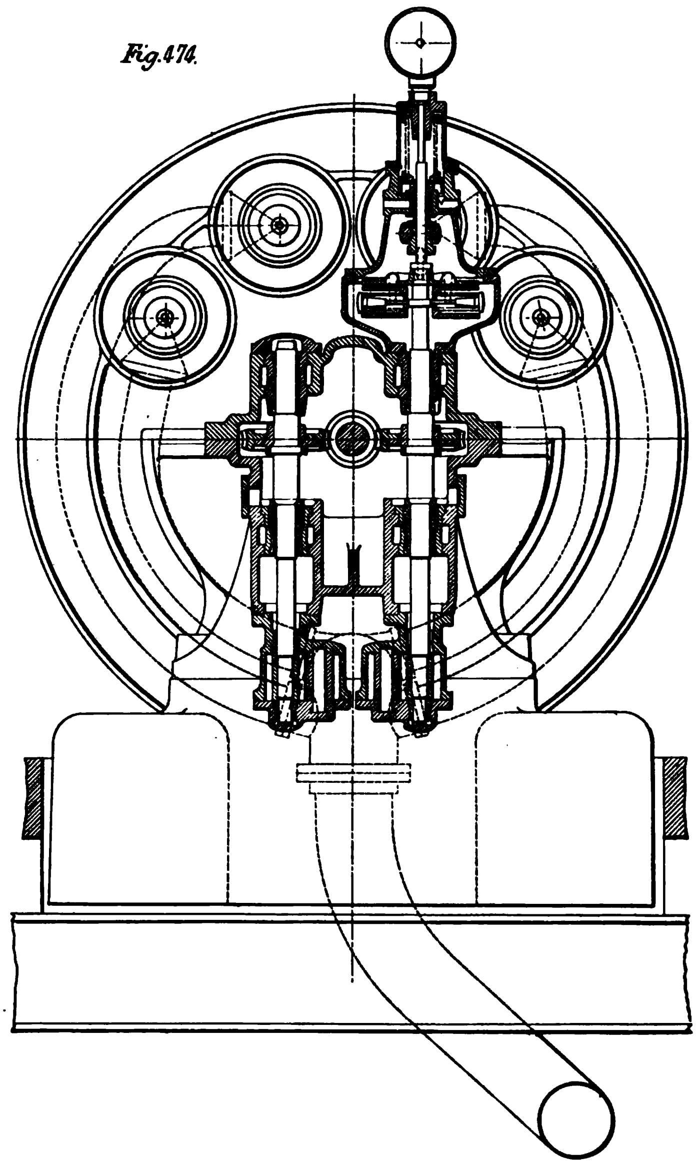
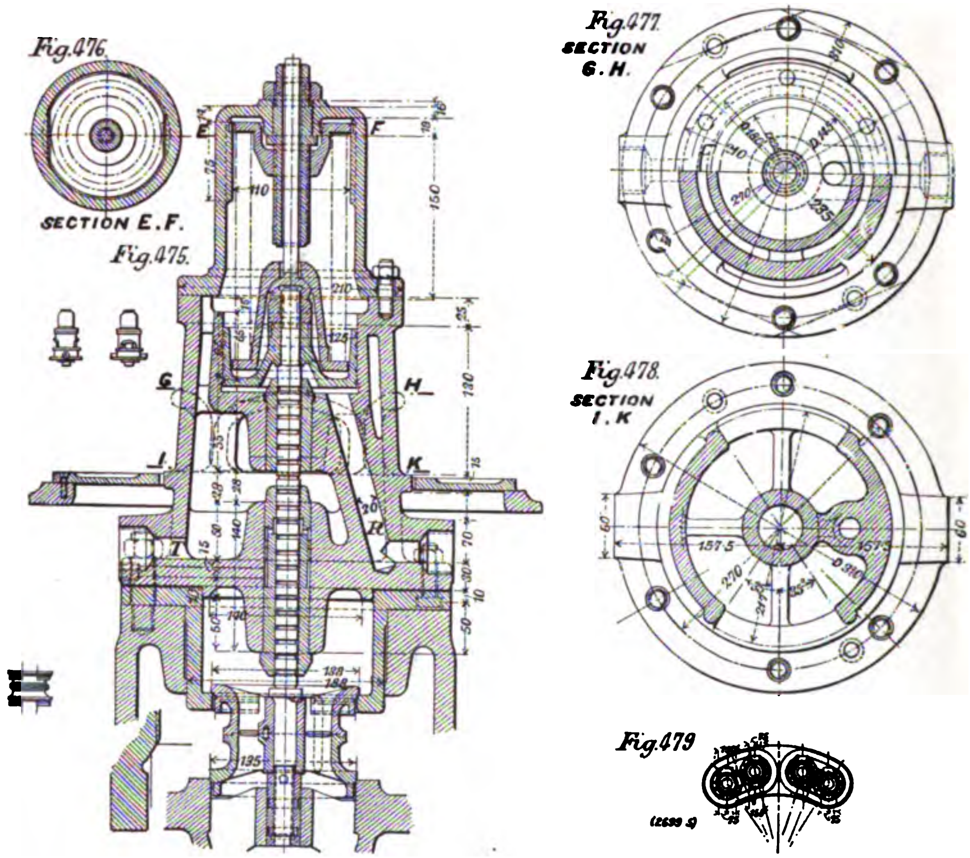


Fig. 474. End View of Disc-and-Drum Turbine.

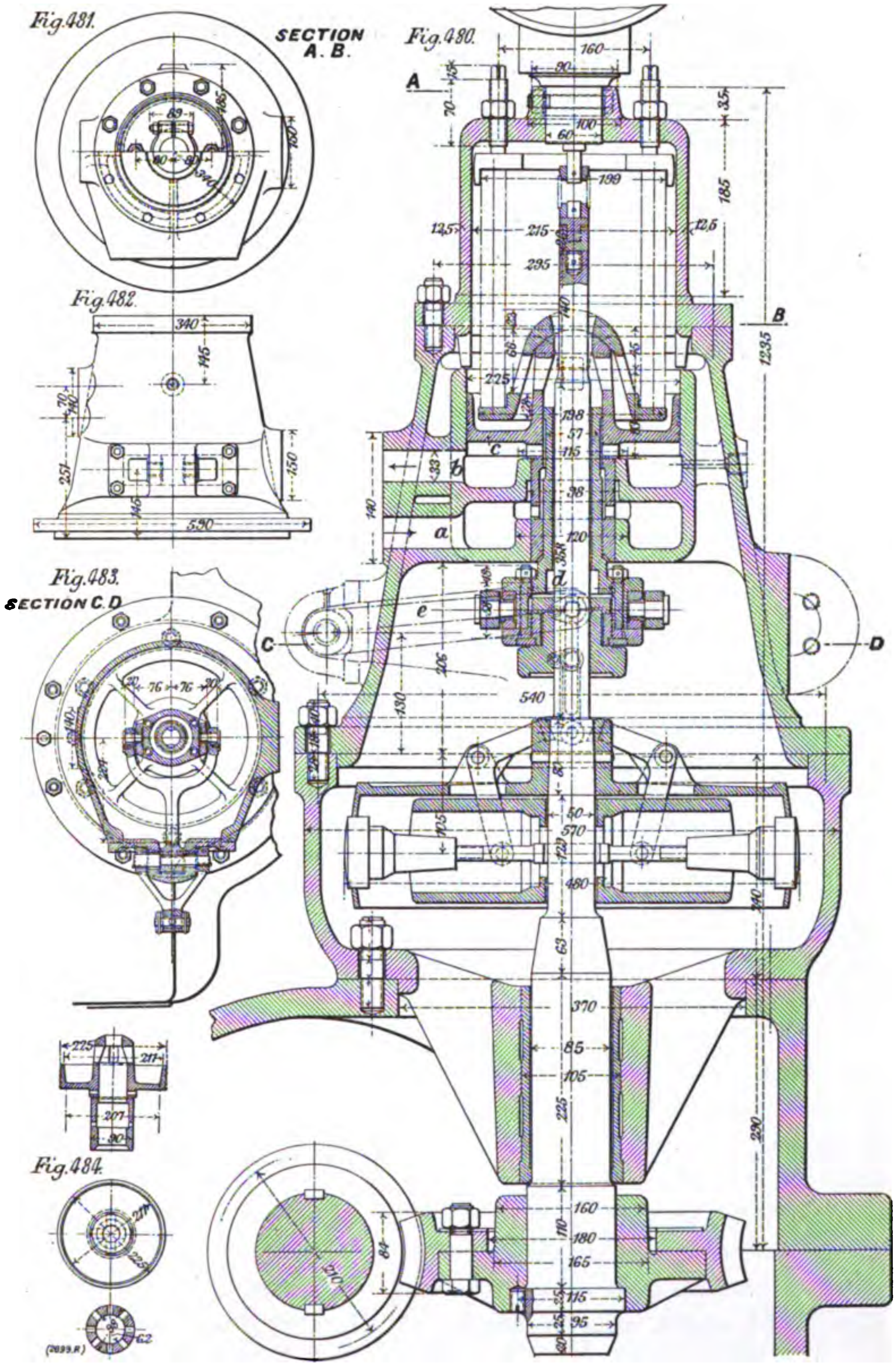
spring already mentioned. These springs are so adjusted that the four valves are operated in succession. Thus, if three are in work, two will be fully open, whilst the third will attend to the governing.

As already mentioned, the oil-supply is controlled by the governor. It enters from the pump by the port *a*, and on reaching the port *b*, puts under pressure the space beneath the piston *c*, raising it against the resistance of the spring above it. If this rise is sufficient, the ports through which the oil passes to *b* will be completely closed; but in the working condition this closure



Figs. 475 to 479. Details of Nozzle Valve.

is only partial, the amount of opening provided being regulated by a cylindrical valve *d*, which is raised or lowered by the governor, according as the speed diminishes or increases. An increase of speed thus tends to throttle the supply of oil to the port *b*. The pressure there is, accordingly, reduced, and the piston *c* is forced down by its spring. This tends to open the ports, and consequently raise the pressure below *c*. There is thus a differential motion between this piston and the valve *d*,



Figs. 480 to 484. Governor for Tosi Turbine.

which has practically the same effect as the "overtaking" motion obtained by means of a floating lever, which is commonly used with steam relays. The lever for the emergency governing device is shown dotted at *e*, Fig. 480, and can also be seen in Fig. 483. Should the speed rise 15 per cent. above the normal, this lever opens a valve on the oil-supply pipe. The pressure there is thus relieved, and the admission valves to the turbine close under the action of their springs.

The steam thrust is for the most part balanced by a dummy, as indicated on the left of the velocity-compounded wheel, Fig. 473, page 334, but any residual thrust is taken by a piston under oil

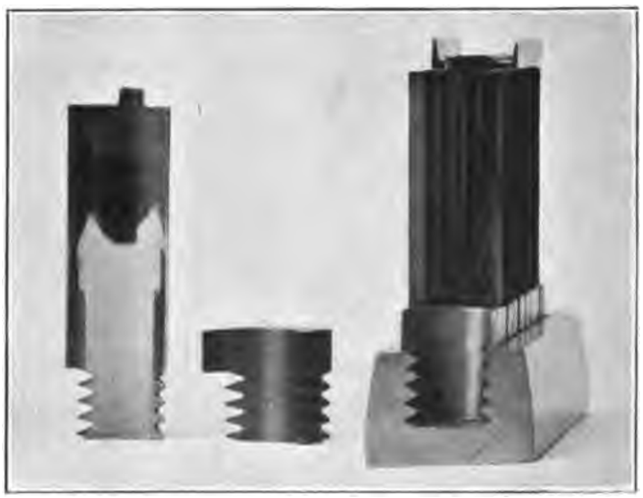


Fig. 485. Blading for Tosi Turbine.

pressure, which, as applied to the Tosi marine turbine, is described in detail on page 348.

The rotor blading is fitted into slots turned on the wheel and the drum. The sides of these slots are grooved, and with these grooves engage teeth machined on the roots of the blades. The arrangement is perhaps best seen in Fig. 485, above, where a blade and distance piece are shown separately. In assembling, these are inserted through "lights" provided in the slots turned on the rotor, as can be seen in Fig. 486. The blades are shrouded, the punched shrouding strip being fitted over snugs machined on the tips of the blades, and secured by riveting over these snugs. The stator blading is of the same pattern, and secured in the same

way to a base block which fits over a dovetail, machined for the purpose in the turbine casing, as best seen to the left of Fig. 489, Plate XXIII., which represents part of the Tosi marine turbine.

Attention may be drawn to the turbine glands, which are of the labyrinth type, but are arranged on sleeves independent and some millimetres clear of the turbine shaft. By this device it is possible to work with very small clearances, since an accidental touch will only heat up this sleeve, and will practically have no

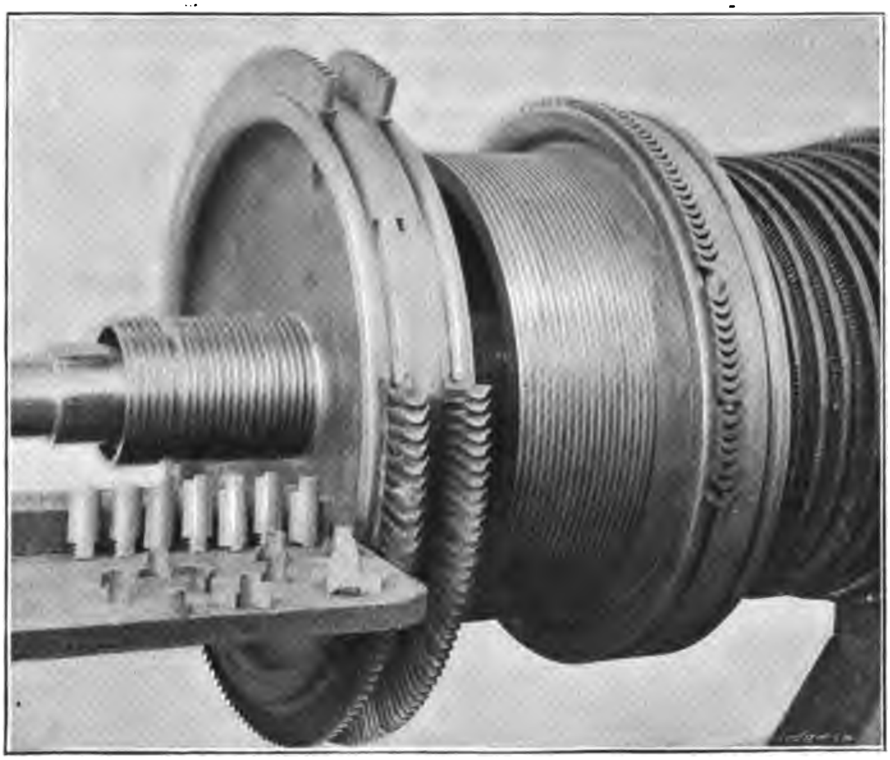


Fig. 486. Blading the Rotor of a Tosi Turbine.

effect upon the shaft proper. The bearings are of the spherical-seated type, as shown in Fig. 473, and are water-cooled.

The condenser is bolted direct on to the turbine casing, without the use of a "concertina" pipe. This is rendered feasible by mounting the condenser on springs, which, whilst supporting its weight, allow it to follow up any changes due to the expansion, by heat, of the casing or of the exhaust pipe.

The condenser is of the counter-current type, and the covers are free from all water connections, so that the tubes are accessible

with a minimum of trouble. The air is extracted by the combination of a steam and of a water ejector, due to Professor Josse. The steam ejector is shown at *f*, Fig. 487. It draws the air from the main condenser and, compressing it to about 3 lb. per sq. in., delivers it into the auxiliary condenser *g*. In this, the steam coming from the ejector *f* is condensed by a jet furnished by the small centrifugal pump *h*. At the top of the auxiliary condenser *g* is a water ejector operated by the circulating water discharged from the main condenser. This draws off all the air in the auxiliary condenser *g*, whilst the condensation here is removed by the centrifugal pump *j*. Tests made by the builders show that very high vacua are realised with this system, and, in fact, the hydraulic ejector in ordinary conditions is itself sufficient to secure an adequate vacuum.

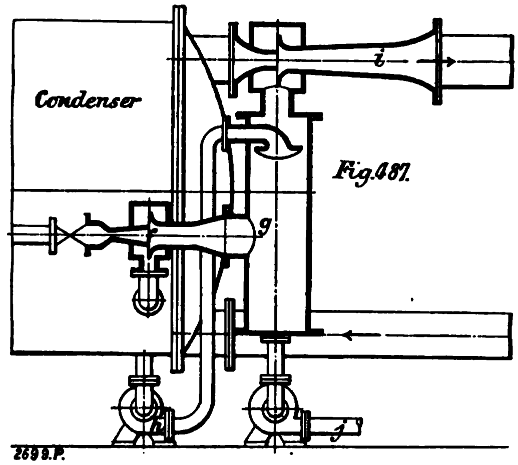


Fig. 487.
Josse Air-Pump for Tosi Turbine.

A marine turbine of the disc-and-drum type, constructed by the same builder, is illustrated in Fig. 488, Plate XXII. This turbine was one of a pair, each developing 7500 shaft horse-power at 600 revolutions per minute, which were fitted to an Italian destroyer. Each turbine drives one of the two screws, and is designed to complete within the one casing the expansion of the steam from an initial pressure of 233 lb. (gauge) down to a 27-in. vacuum. In the same casing a reverse turbine is fitted, as shown to the right in Fig. 488. This reverse turbine develops 3450 shaft horse-power at 400 revolutions per minute. As will be seen, both the main and the reverse turbines are of the "mixed" type. The main turbine comprises six velocity-compounded stages followed by a drum carrying fourteen rows of reaction blading. The wheel in the first stage has four rows of moving blades, and the other wheels three rows each. The mean diameter of this impulse section of the turbine is 60 in. The blading of the first wheel and of the second is represented to an enlarged scale in Fig. 489, Plate XXIII., and, as shown in Fig. 490, the

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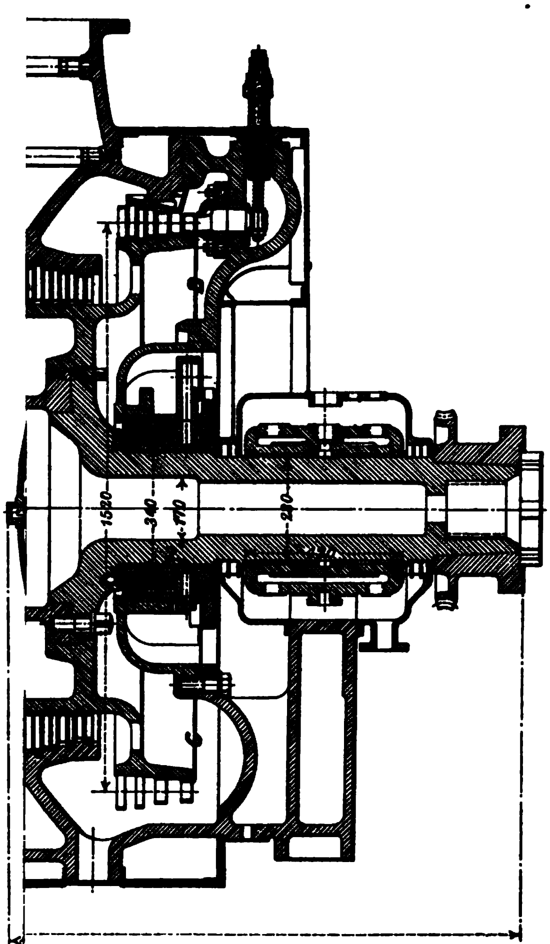

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blading for the last of these velocity-compounded stages is mounted on the forward end of the drum.

The turbine casing is of cast iron, and is made in four parts, rigidly bolted together. Of these four parts, two constitute the casing for the high-pressure section, whilst the other two accommodate the reaction drum and the whole of the astern turbine. The casing is strongly ribbed, and the flange bolts located as closely as possible to the shell, so as to reduce to a minimum any possibility of distortion. The risk of this is, moreover, much minimised by the large diameter and relatively short length of the turbine. The high-pressure casing has turned internal seats to take the nozzle castings, and the latter in their turn are machined to form seats for the diaphragms, which are secured in place by bolts which pass, as shown in Fig. 489, through holes formed half in the nozzle castings and half in the diaphragms themselves. The fixed guide blades are mounted in castings turned to slide on to dovetailed seatings machined for them in the main casing, as shown in Figs. 489 and 490, Plate XXIII.

The feet by which the casing rests on the seatings on shipboard are low, so as to reduce the leverage between these seatings and the centre line of the turbine. The after end of the casing is securely fixed against longitudinal movement, but provision is made for the free expansion of the casing transversely. At the forward end the feet can slide both longitudinally and transversely. The exhaust opening is well stayed. As shown in Fig. 488, Plate XXII., drains are provided to keep clear of water the six impulse compartments. In order to ensure an absence of casting strains in the casing, the latter was, after rough machining, carefully annealed at a high temperature, and on final assembling it was tested to a pressure of 323 lb. per sq. in.

The general construction of the rotor is clearly illustrated in Fig. 488, Plate XXII., and some details are shown to a larger scale in Figs. 489 and 490, Plate XXIII. It will be seen that it consists essentially of a very stiff hollow shaft, made in two parts, which are bolted together near the right-hand end, as indicated. The joint, which is shown separately in Fig. 492, Plate XXIII., is thus close to a bearing, and is consequently under but a very small bending stress. The drum is shrunk on to the shaft

just at the joint, and is secured from turning by means of set bolts, screwed half into the drum and half into the joint flange, to act as keys. At its forward end the drum is simply supported from the shaft by means of a somewhat thin diaphragm, as illustrated, and this is sufficiently elastic to allow of slight differences in the expansion of the drum and the shaft. Like the casing, the shaft and drum are both thoroughly annealed after rough machining.

The wheels are mild-steel forgings, having, as shown in Fig. 489, large openings in their discs, to secure lightness, and also to equalise the pressure on both sides. They are forced fits on their seats and are prevented from turning by means of two keys "on the flat," one of which is represented at the top of the shaft in Fig. 491. The whole series of wheels are secured against longitudinal motion by a clamping nut at the forward end, which is clearly shown in Fig. 488, and to a much larger scale in Fig. 489. This nut is, moreover, shown also at O, in Fig. 493, annexed. It is prevented from unscrewing by means of a cheese-headed machine screw, which is itself locked in place by caulking the surrounding metal into its slot. As shown in Fig. 489, the fins of the diaphragm packings do not rest directly on the wheel bosses, but on special sleeves standing some millimetres clear of them, and secured to the wheel centres by bolts. Another view of these sleeves is represented in Fig. 491, Plate XXIII. As they stand clear of the shaft, the latter cannot be warped by unequal heating, should excessive friction arise between the sleeve and the diaphragm. The rotor, as a whole, has a critical speed sufficiently in excess of its running speed to render it thoroughly safe against any danger of whipping. The wheels, the drum, and the shaft are all balanced separately before final assembling, and are then again re-balanced, being subsequently run for an hour or so

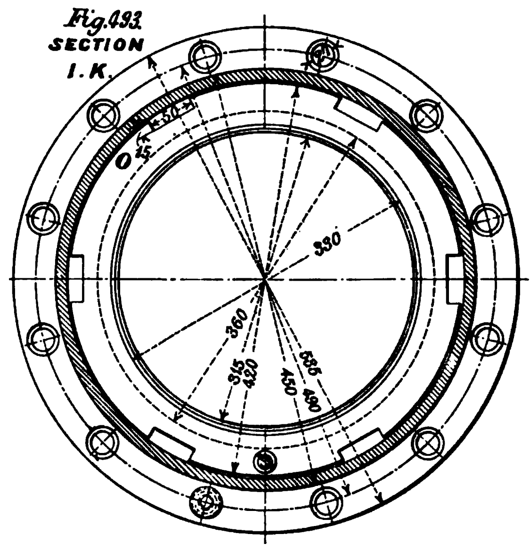


Fig. 493. Nut for Securing Wheels on Shaft.

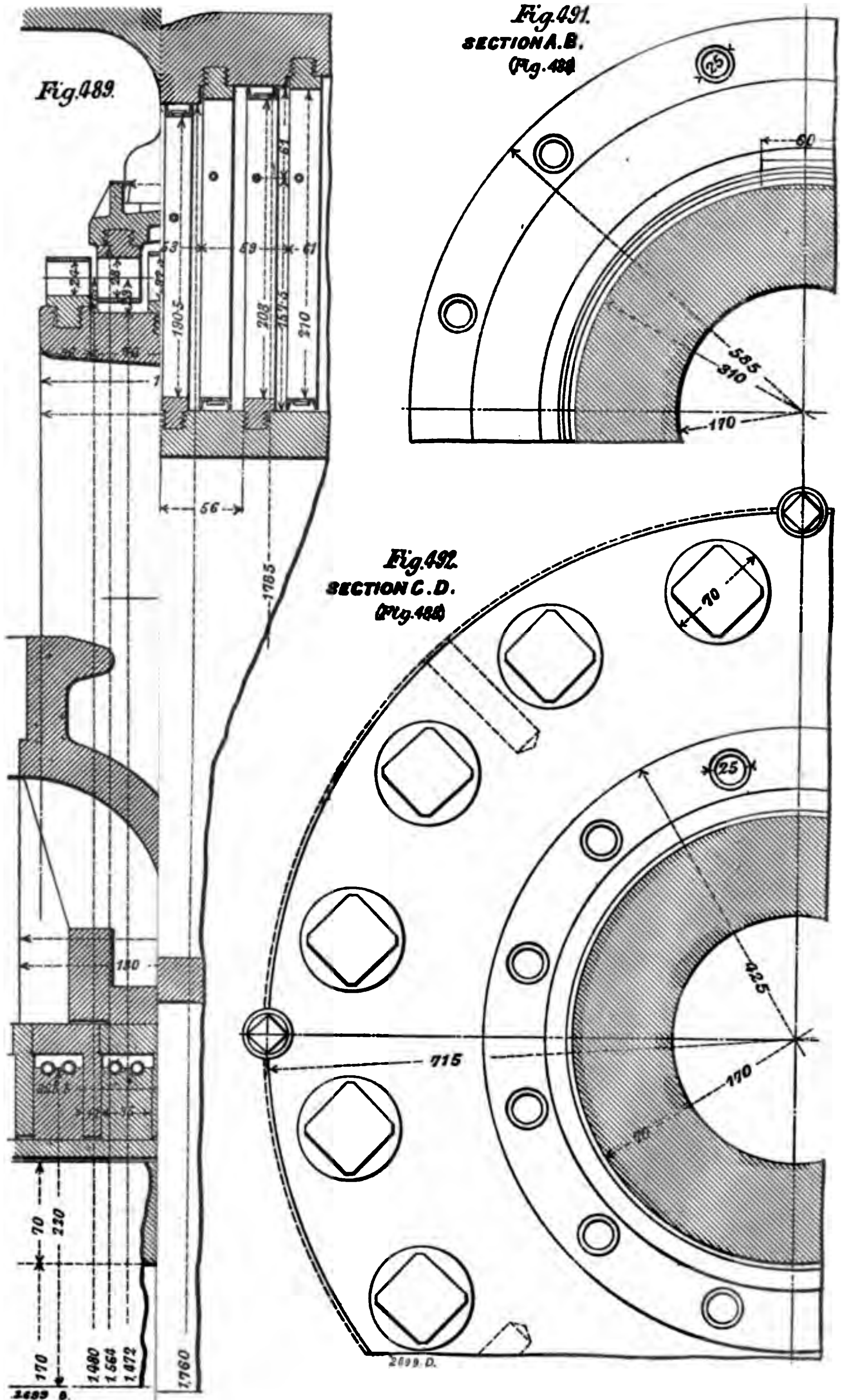


Fig. 489.
Fig. 491.
SECTION A.B.
(Fig. 488)
Fig. 492.
SECTION C.D.
(Fig. 488)
1785
1760
715
425
170
585
310
70
25
60
56
210
190·5
2489 B.
2889 D.

To Face Pag

at 700 revolutions per minute, followed by a short run at 800 revolutions. During this final test the turbine is supplied with steam superheated to 300 deg., and operated with a low vacuum, so as to ensure that the temperature shall everywhere attain limits in excess of what will be reached in subsequent operations on ship-board.

The diaphragms, as shown in Figs. 489 and 490, have a composite structure, consisting of an outer ring of cast iron, into which are cast the nickel-steel partitions which form the guide

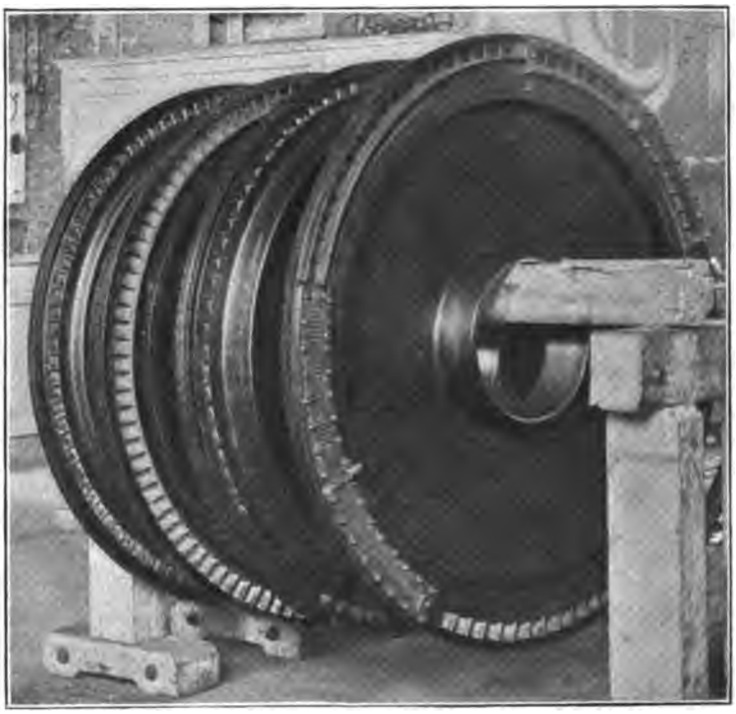


Fig. 494. Diaphragms for Tosi Turbine.

blades. The central portion of the diaphragm, on the other hand, is made of forged steel, so as to ensure lightness, and, as already mentioned, is secured to the cast-iron outer ring by bolts lying half in this ring and half in the diaphragm. At its centre hole each diaphragm is fitted with a series of brass fins, the inner edges of which are reduced to a fine knife-edge, so that if accidentally touched by the adjacent rotating surface, there is no serious development of heat, which may lead to a distortion of the shaft. An additional safeguard has, as stated above, been provided in the present instance,

since these fins can in no case come into contact with the shaft proper, but only with a sleeve standing clear of the latter, and bolted to the wheel bosses. A set of diaphragms ready for erection is illustrated in Fig. 494, page 343. No diaphragm is, of course,

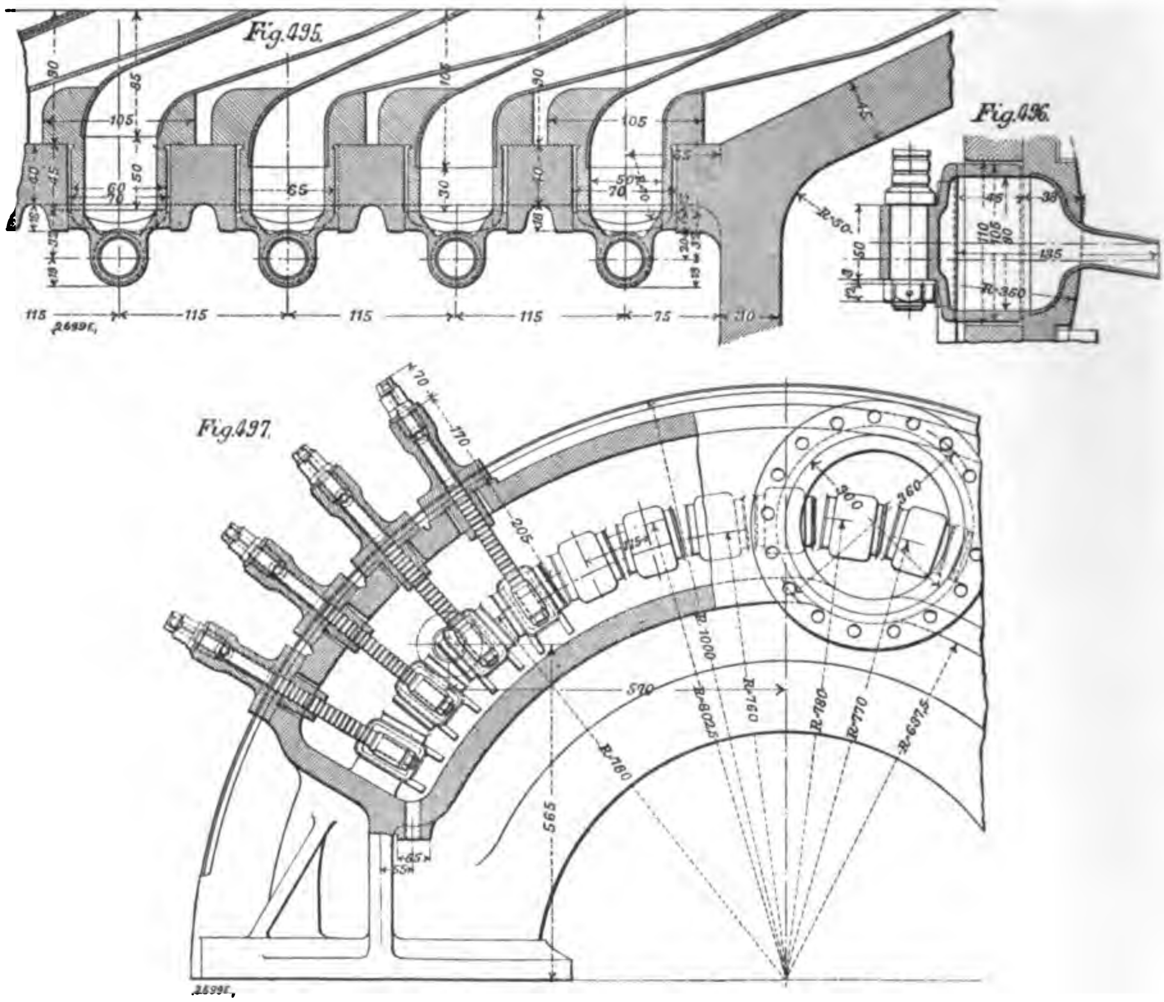
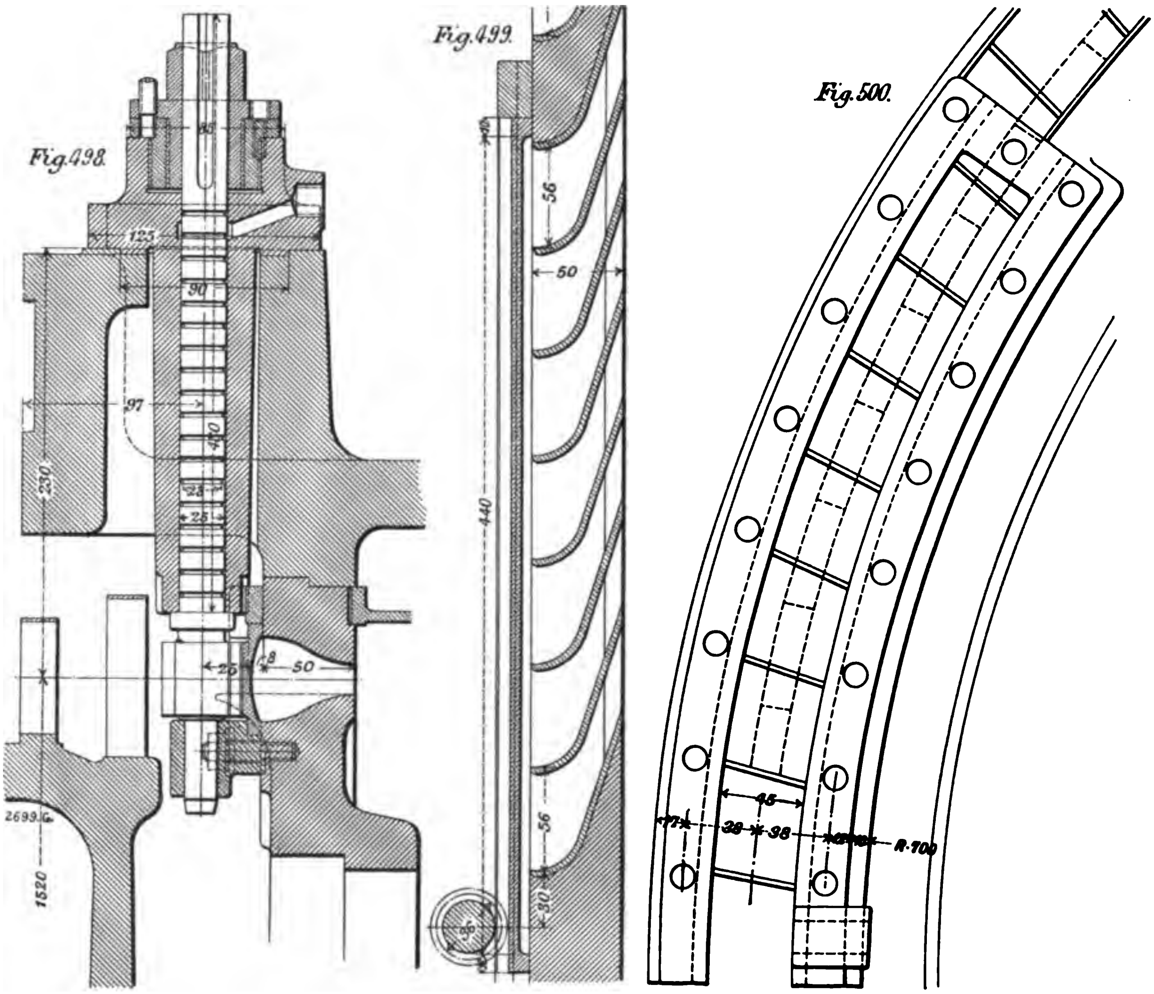


Fig. 495 to 497. Nozzles for Tosi Turbine.

needed for the first wheel, but merely a nozzle box, of which details are shown in Figs. 495 to 497. In this nozzle box there are fifteen nozzles, each controllable by an independent valve. Of this total, however, only eleven are in use at full speed, when the power to be developed and the total steam to be passed is very much greater than at cruising speeds. To get this large volume of steam through the turbine, the pressure in the first

compartment must be much greater than suffices to pass the smaller weight traversing the turbine at low speeds. Hence, if the same nozzles were used in both conditions, they would have to be either too divergent for economy at full load, or insufficiently divergent for economy in cruising conditions. Hence at cruising



Figs. 498 to 500. Cut-off Valve for Diaphragm No. 1.

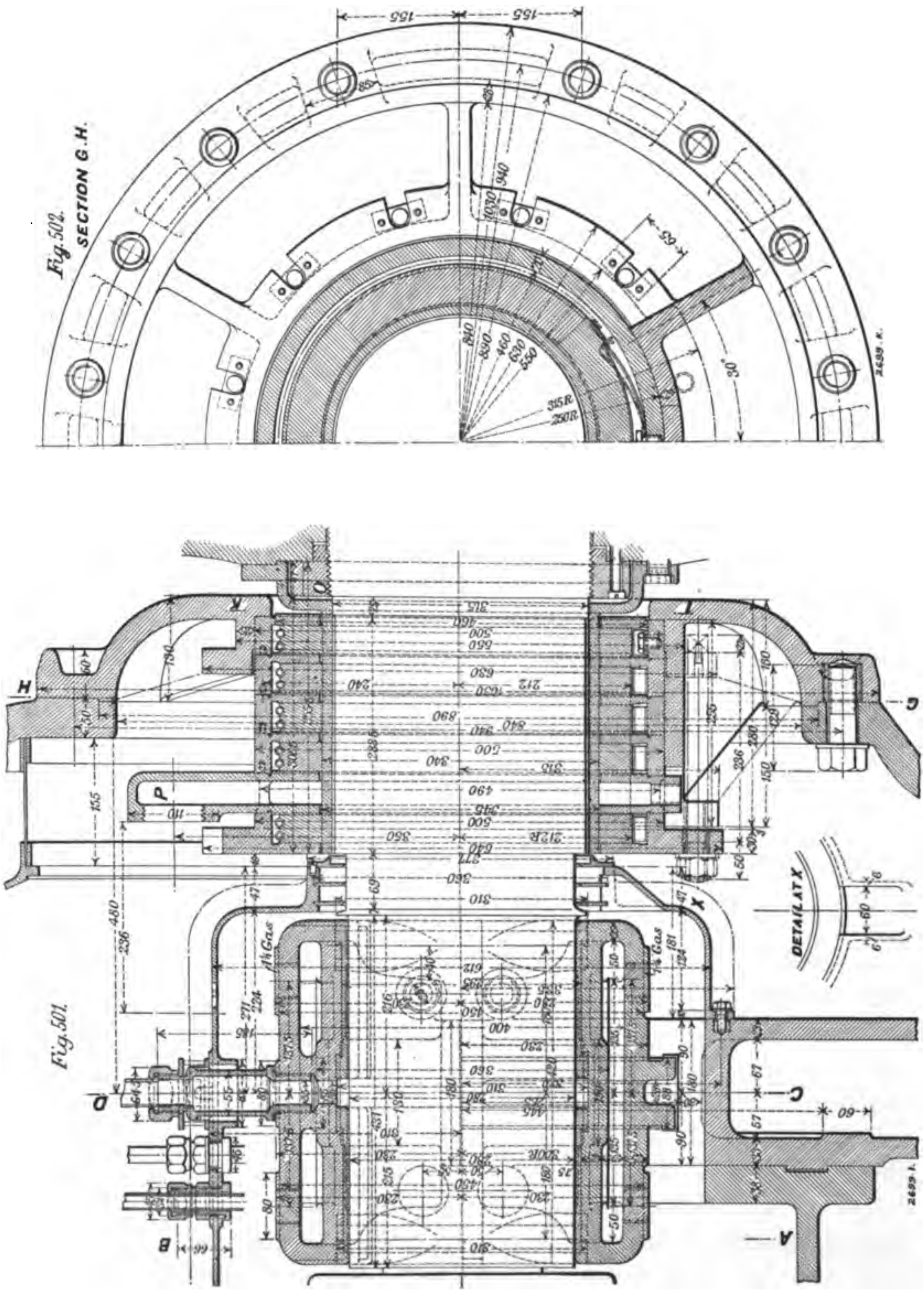
speeds the main nozzles are completely closed, and the steam enters the turbine through four special cruising nozzles. The difference between these and the nozzles used at full load is clearly indicated in Fig. 495, where the widely-flaring nozzles shown on the right are those used at low powers, whilst the nearly parallel nozzle on the left is one of those used at full speed. Still further to augment

the economy, provision is made for shutting off some of the nozzles in the first diaphragm by means of a circular sluice valve operated by rack and pinion. This arrangement is clearly shown in Figs. 498 to 500, and in the first of the set of diaphragms shown in Fig. 494. As will be seen from Fig. 499, the blading in these intermediate diaphragms is all of the non-divergent type, the pressure drop being for these stages below the critical value.

The rotor blading is of the same type, and secured in the same way as in the case of the turbine for generator driving already described.

The glands are made tight by carbon packing rings, the whole arrangement being clearly shown in Figs. 501 and 502, on page 347. These rings are each made in four parts, and are held in place by springs, as indicated. To reduce wear, they are, moreover, supported from below by coach springs, being cut away for this purpose, as indicated in Fig. 504. In the case of the high-pressure gland, a "leak-off" is provided, as shown at P in Fig. 501, to prevent the leakage escaping into the engine room. A similar arrangement is used in the case of the low-pressure gland to supply packing steam. A special feature of this packing is that the rings do not bear directly on the shaft, but on an independent sleeve, which is bolted to the hub of the first wheel, as shown in Fig. 489, Plate XXIII., and serves also, as can be seen in Fig. 493 and in Fig. 501, to cover the nut O, by which the wheels are clamped into place. Since this sleeve stands quite clear of the shaft, any non-uniform heating of the gland arising, say, from the accidental presence of foreign bodies, cannot give rise to distortion. At the worst the sleeve may be damaged and the steam leakage increased until the sleeve can be replaced by a new one, an operation which is easily effected. One of the carbon rings is shown in Fig. 504.

The various components of the gland are held in place by long bolts. These are secured at their inner ends by cotters, as indicated in Fig. 501. The main bearings of the turbine are kept as close to the gland as possible, so as to shorten the length of the rotor, and thus increase its stiffness. As shown in Fig. 501, they are water-cooled. They have cast-iron bodies, lined with white metal. The lubrication is "forced." An oil thrower, shown on the right in Fig. 501, checks the creeping of oil, and to the



Figs. 501 and 502. Gland and Main Bearing.

right of this again wipers are provided, so as to further guard against the entrance of oil into the gland, and thence to the condenser.

A special feature of the turbine is the method of taking up any difference between the steam thrust forcing the rotor astern and the propeller thrust in the opposite direction. At full power this residual thrust amounts to about 20 per cent. of the maximum thrust, but at cruising speeds the rotor is very nearly balanced. This thrust is taken by oil pressure, exerted on a piston fixed on

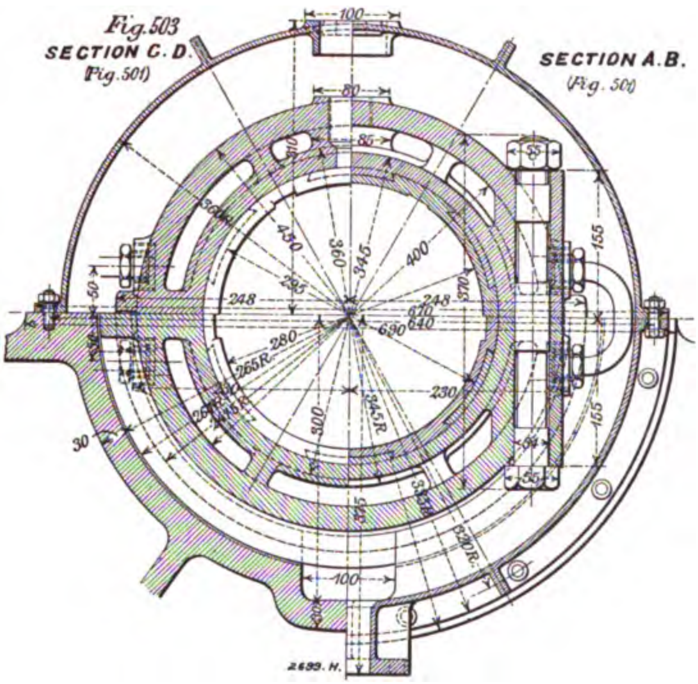


Fig. 503. Main Bearing.

Fig. 504. Carbon Packing Ring.

the left-hand end of the shaft, as represented in Fig. 488, and in greater detail in Fig. 505, page 349. This piston is represented by Q, and, it will be seen, is packed by means of a labyrinth. Oil under pressure is constantly forced into the chambers on both sides of this piston, and has to escape between the collars of the thrust blocks in each, shown on each side of the piston. If the thrust is in one direction, then the distance between the collars of the one block is diminished, whilst the clearance at the other block is increased. As a consequence the oil leaks away more easily from one chamber than the other, and difference of pressure is thus

established between the two faces of this piston. The latter, therefore, moves so as to diminish the clearance on the one side and

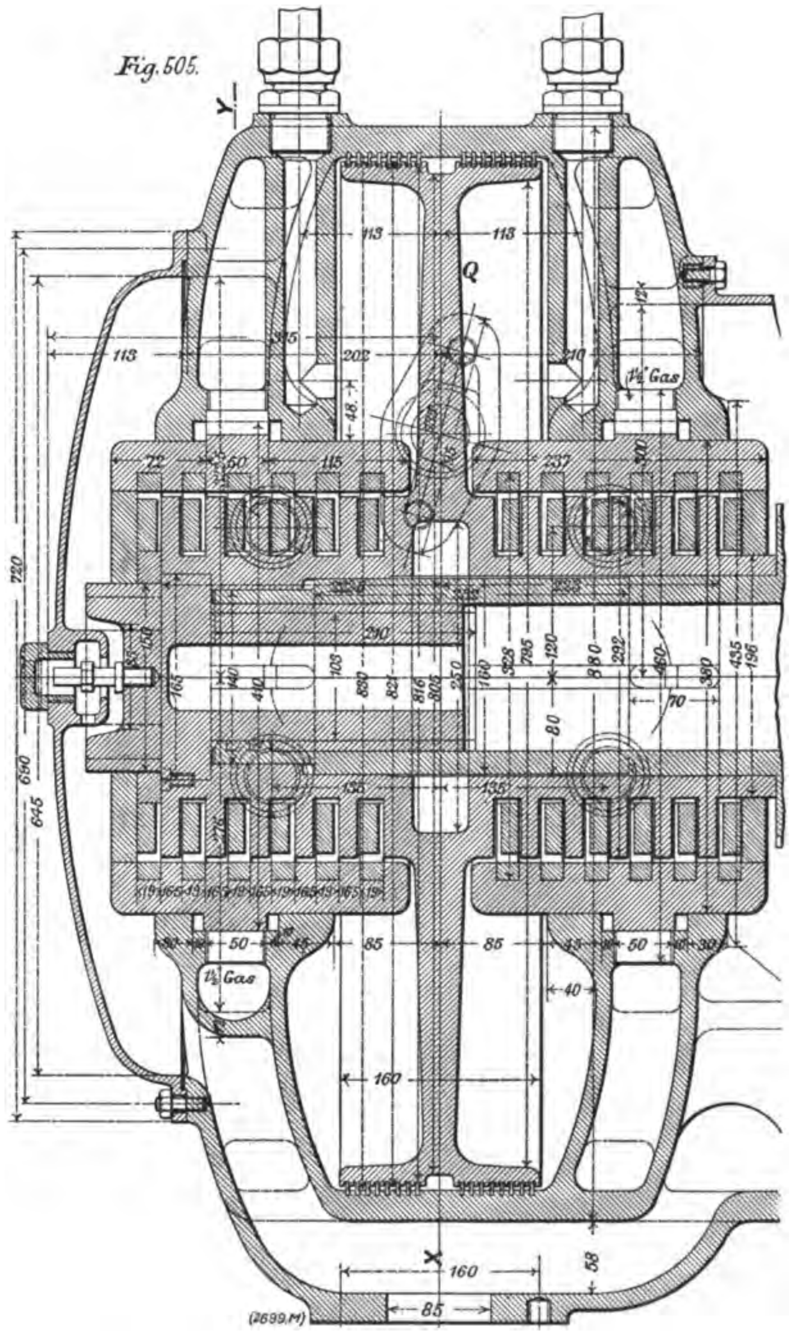


Fig. 505. Thrust Block for Tosi Turbine.

increase it on the other, thus automatically preventing any contact between the collars of either thrust block, which accordingly remain

free from wear indefinitely. A bastard section through one of the thrust blocks is shown to a smaller scale in Fig. 506. The block is, it will be seen, water-cooled. Very complete workshop tests were made of this thrust block before it was finally fitted on shipboard.

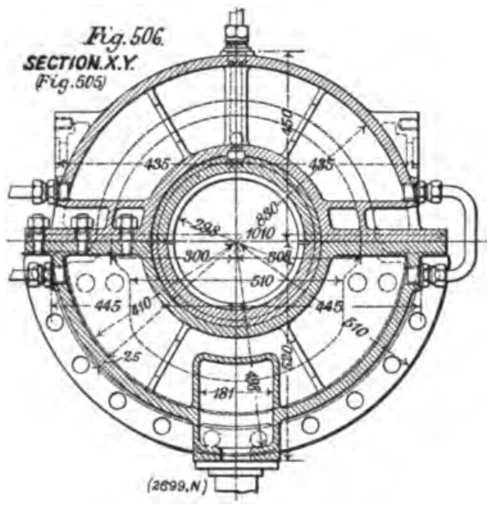


Fig. 506. Section through Thrust Block.

The oil for this block and for the bearings is supplied by an independent steam-driven oil pump. The discharge from the thrust block and bearings is collected and filtered, and is then drawn through a cooler, and pumped again into the bearings.

CHAPTER XXX.

THE LJUNGSTRÖM STEAM TURBINE.

IN the case of a water turbine the volume of water delivered into the tail race is identical with that which enters the inlet pipe to the turbine, but in the case of a steam turbine there is an enormous increase in the volume between the stop valve and the exhaust branch.

It was early obvious that a compound radial-flow machine would accommodate itself somewhat more naturally to this change of volume than a turbine of the parallel-flow type. Admitting the high-pressure steam near the centre of such a turbine the area available for flow is proportional to the radius of the guide-blade circle, if the blade height be constant, and increases therefore with each successive row of blades.

A number of radial-flow turbines were accordingly constructed in the early days by Sir C. A. Parsons and his associates, but the difficulty of reducing tip leakage to a reasonable figure was such as to lead to a reversion to the parallel-flow type of turbine at a subsequent date.

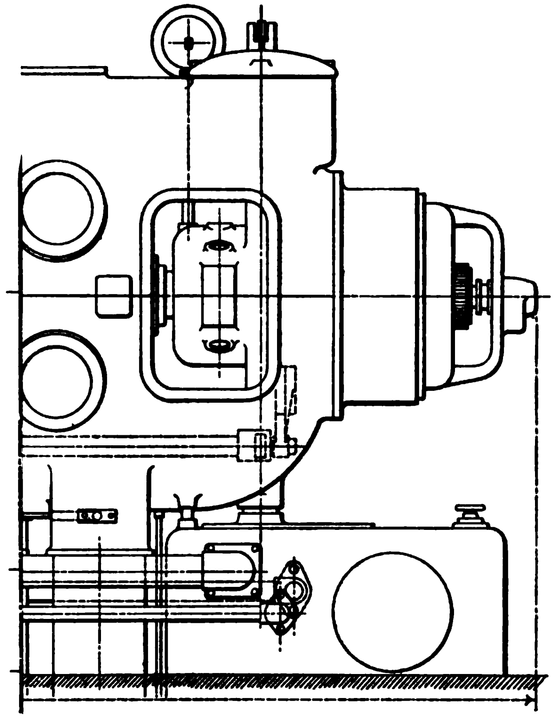
One of the prime difficulties encountered in steam-turbine design is to secure a favourable ratio between the steam speed and the blade speed. In reaction turbines in particular the blade speed is commonly only a fraction of its theoretical best value, and even so the number of rows of blades required is generally very large. Here, again, it was early observed that with the radial type of construction it would be quite possible to run what, in the case of an ordinary parallel-flow turbine, would be the guide blades in a reverse direction to the ordinary rotor, thus doubling the relative speeds of the two sets of blades and diminishing the number, required to get a stated efficiency, to a quarter of what would be necessary for a "single-rotation" turbine. Here again, owing to leakage difficulties,

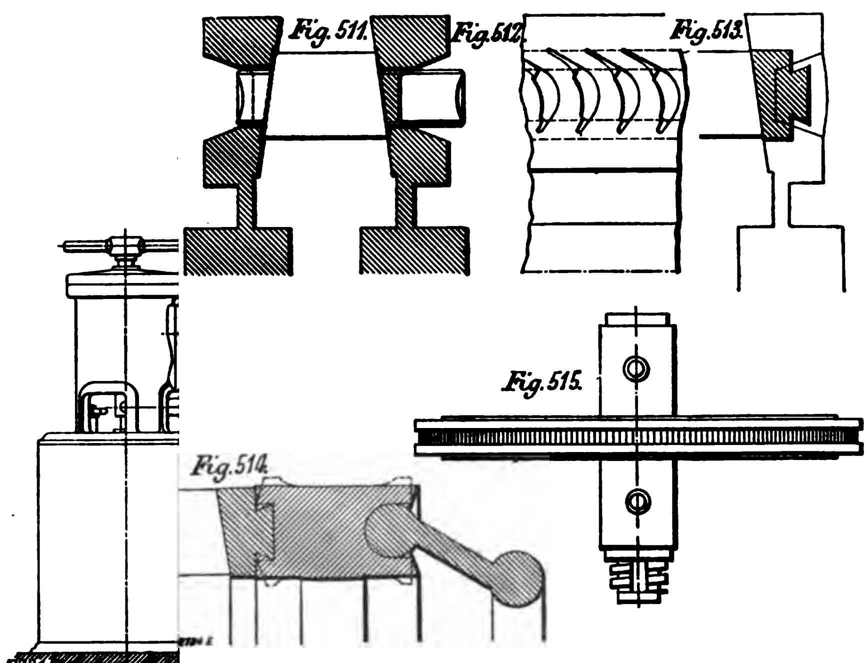
early experiments in this direction gave unsatisfactory results. Later on the problem of designing a turbine of this character was taken up by Mr. Birger Ljungström, of Liljeholmen, Sweden, who succeeded in practically suppressing the leakage difficulties, and constructed a turbine which, though of 1000 kw. output only, surpassed all previous records in steam economy.

A general arrangement of the 1000-kw. turbo-generator, designed to run at 3000 revolutions per minute, is shown in Fig. 507, Plate XXIV. The turbine is in the centre, with an alternator on each side of it, and at the end of one of the alternator shafts is mounted a small exciter which supplies the fields of both machines. The lower half of the turbine casing is formed into an exhaust branch, which is bolted directly to the condenser beneath, without the intervention of any expansion joint or similar device. The exhaust branch is the only rigid support the turbo-generator possesses, so that the latter is as free as possible from any strain due to its attachments. Springs, contained in cast-iron boxes beneath each generator, take the weight of these, and prevent strains due to their overhang.

The over-all length of the turbine is only 17 ft. $4\frac{1}{16}$ in., the greatest diameter 4 ft. $2\frac{5}{8}$ in., and the height to the top of the governor is about 5 ft. 6 in. Fig. 507 shows the arrangement and comparative accessibility of all bearings, the air passages through the generators, the branched steam connection to the turbine proper, and particularly the exceedingly small dimensions of the latter. The end view, Fig. 508, shows the emergency trip lever and its cable connection *a* between the stop valve and the turbine casing. This cable is under tension, and holds shut an oil-relief valve in the relay by which the stop valve is operated. It can be released either automatically by the melting of a fusible plug in the bearings or by hand. It should be noted that the whole of the turbine is surrounded by exhaust space, and thus there is no need for lagging of any kind upon the surface of the casing, the temperature of which never exceeds that of the condenser. This arrangement not only minimises radiation losses and permits of ready access to the bolts, but conduces to a pleasant temperature in the neighbourhood of the machine.

While Fig. 507 makes clear the relative position of the various





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parts inside the casing, it shows nothing as to the actual construction of the turbine itself, beyond indicating the very small space it occupies in the centre of the plant.

Steam, after leaving the stop valve, enters the casing by a pipe passing through the side of the exhaust branch, where it divides and ascends to the turbine by means of two vertical pipes *b*, *b*, as in Fig. 507. These pipes are of steel, and radiation from them is minimised by jackets of thin steel, enclosing an air space. This branched pipe is made by welding, and its jackets are also welded on. The ends of the branch pipes are shown at A in Fig. 516, page 354. They are turned down on the outside, leaving a shallow collar about $\frac{1}{2}$ in. wide near the end, this collar being a tight fit into a thin steel bush firmly fixed into the cast-steel "side discs" of the turbine. These side discs fit in turned seatings in the casing, and serve to locate the turbine. When the turbine is removed it is lifted off the ends of the branch pipes. The joint at this part is stated to have proved perfectly steam-tight. Except around the collar, there is no contact between the pipe and the bush, but steam freely circulates between them.

The steam from each pipe enters an annular chamber B (see Fig. 516, page 354) in the side discs, whence it passes by means of a number of large holes C to the space between the turbine discs. The opposing faces of the discs D carry rings of blading, supported from the discs by means of conical rings of bull-headed section, which will be described later. On emerging from the last blade ring, the steam enters a diffuser E, comprising a number of diverging channels, in which the residual steam velocity is re-converted into pressure, and the vacuum at the blade edges is thus slightly increased above that maintained in the condenser. This diffuser is built up of sheet steel by welding, and is mounted on suitable supports in the turbine casing. It comes away with the turbine when the latter is removed.

It will be at once understood that in such a turbine steam packings have to be provided both to prevent the steam escaping along the shafts where the latter pass through the side discs, and also to prevent it escaping freely up the back of the running discs, and thus passing uselessly into the condenser. The packing used for this latter purpose has also to fulfil another function—namely,

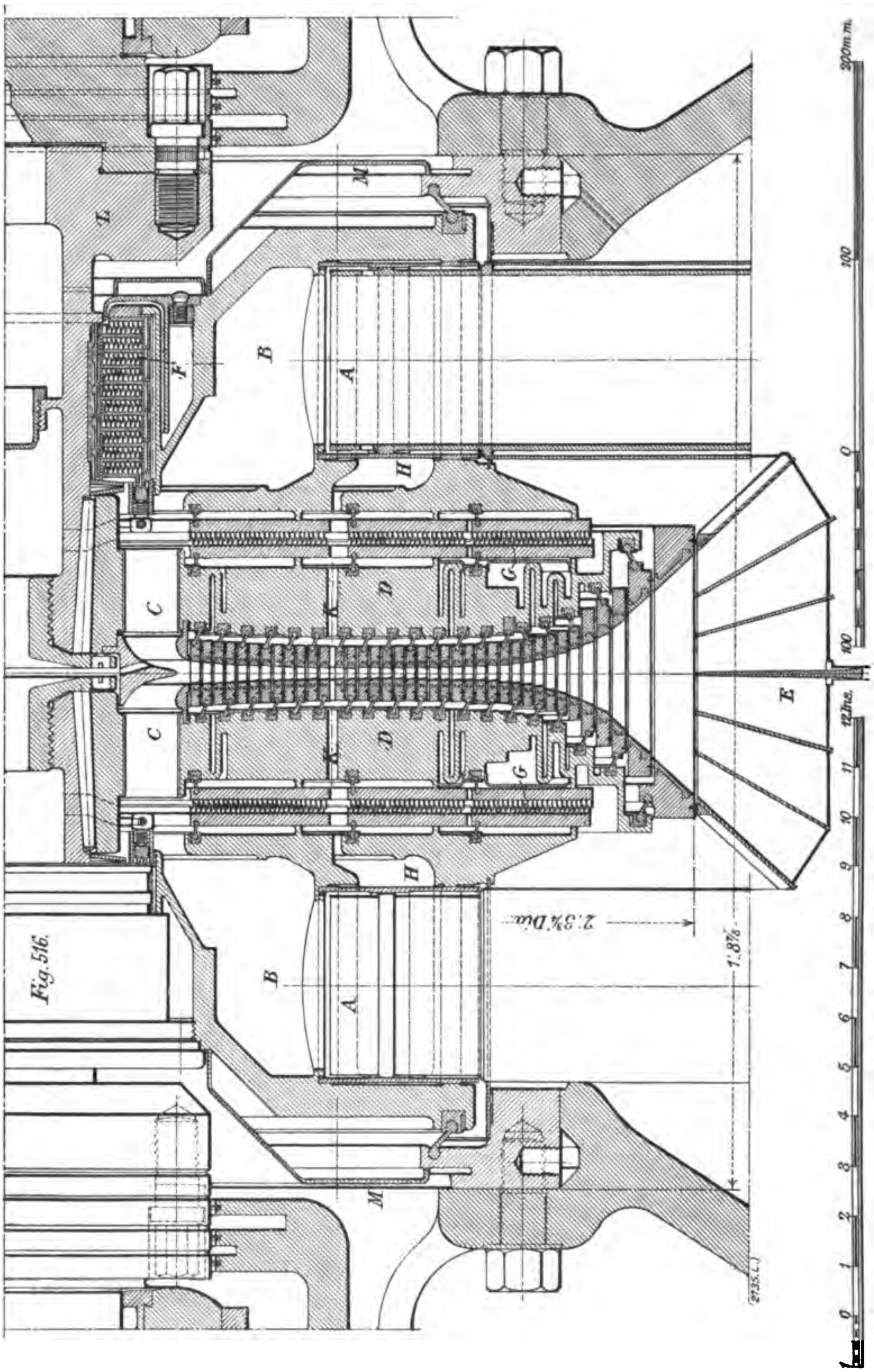


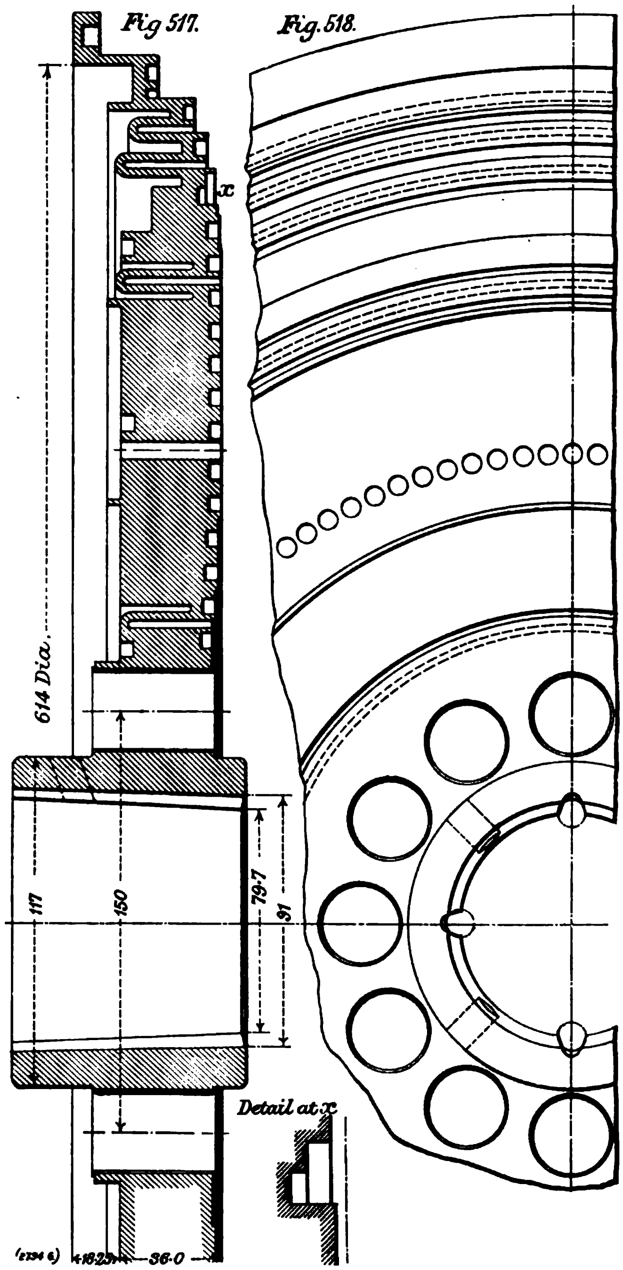
Fig. 516. 1000-Kilowatt Ljungström Steam Turbine.

to constitute a balancing device to neutralise the pressure of the working steam tending to force the discs apart. The type of packing adopted for this purpose is shown at G, Fig. 516, and to a larger scale in Figs. 509 and 510, Plate XXIV. The shaft packing is marked F in Fig. 516, and with the dummy packings will be described in detail later. When the turbine is required to take an overload, steam is admitted to the annular chambers H by means of either or both of the overload valves on the casing. It passes through the turbine discs by the holes K, and so enters the blading.

Each turbine shaft L, Fig. 516, is hollow, and carries the running part of the packing F and the turbine discs D. The packing is mounted on solid feathers, and held in position against a collar by the turbine disc, between which and the packing are two concave spring washers. The disc is keyed to the shaft by a number of taper pins, and is held in place by a flanged spigot screwed into the end of the shaft. The turbine shafts themselves have no bearings, but are registered and fastened with set screws each to its own generator shaft, the bearings belonging to the later. There is, however, very little overhang, as each turbine shaft is only about $10\frac{1}{2}$ in. long. The weight of the two running discs, complete with blading, being only 265 lb. and 303 lb. respectively, no difficulty is experienced in securing true running.

As shown in Fig. 516, the turbine is of very small dimensions for its power, measuring but $27\frac{3}{4}$ in. in diameter by $20\frac{7}{8}$ in. long. Note should also be made of the extreme care taken at every point to minimise heat losses, and to avoid possible trouble from the very high superheats for which the turbine is designed. The only external parts of the turbine which are heated by live steam are the small surfaces of the side discs, and these are lagged and cased with sheet metal, as shown at M in Fig. 516. Every part is circular, and without flanges or horizontal joints. The packings, dummies, turbine discs and blading system are all of nickel steel, and wherever there is either a gradual or an abrupt temperature gradient steps are taken to provide against distortion. This is particularly noticeable in the formation of the turbine discs, which, as best seen in Figs. 517 and 518, page 356, are nearly cut through in places by deep annular grooves, to avoid the distortion which

would inevitably otherwise occur in thick discs highly heated at the centre and comparatively cool at the circumference. The most elaborate provision against distortion, however, is the system of conical rings forming a connection between parts which are liable to be momentarily or permanently of different temperature. The side discs, which necessarily become hot when the turbine is at work, are also held to their supporting rings, which are cold, by this device. The two sets of the dummy discs are similarly supported, one to a facing on its side disc, and the other to the back of its turbine disc. Furthermore, every individual blade ring is fixed to one or other of the turbine discs by the same method of attachment, which will be described in detail later on.



Figs. 517 and 518. Disc for Ljungström Turbine.

The blading of the Ljungström turbine consists of a number of concentric blade rings carried alternately by the two turbine discs as shown in Fig. 516, on page 354. A section through a blade ring, 2.4 times full size, is given in Fig. 509, on Plate XXIV. In the 1000-kw.

turbine there are altogether thirty-eight rings. The blades themselves are of the Parsons pattern. Reckoning from the centre, the first thirty-three rows are 5 mm. wide, the next two are 7 mm. wide, and the last two 12 mm. and 20 mm. wide respectively. The blading is first manufactured in lengths of about 3 ft., being milled from solid round bars of nickel steel. This is done on an ordinary horizontal milling machine, having four formed cutters mounted on its spindle. The first cutter roughs out the convex side of the blade, the next finishes this side, and the third and fourth respectively rough out and finish the hollow side. Although the blade strip is extremely well finished when it leaves the machine, Mr. Ljungström has it polished to a mirror-like surface, both inside and out, before it is built into the rings. Polishing is effected by the use of a simple machine using a tape band about 2½ in. wide, treated with polishing paste.

The strip is next cut into blade lengths, which are notched at the ends for insertion into the rings which hold them. The shape of such a blade is seen in Figs. 511 and 512, Plate XXIV., in which it is shown with the ends inserted into two discs, from which the rings forming the blade roots will afterwards be turned. These discs, which are of soft iron, are mounted on a mandril and their edges turned, inside and out, to the contour shown in Fig. 511. They are then dismounted and punched to receive the ends of the blading, a second punching process nicking the inner faces of the discs in such a way as to form registers for holding the blades to the exact angle. When the punching is done, the discs are reassembled on a mandril with the blades in position, as shown in Fig. 515. The mandril is then held vertically, and the rim of the disc and projecting blade roots are welded up solid together, by melting iron wire into the groove by the aid of an oxy-acetylene blow lamp, the groove being filled up with the molten metal as indicated in Fig. 513, Plate XXIV. Both ends of the blades having been welded into the discs by this process, the mandril is again put in the lathe, and the discs and welded part are turned down and the ring parted off. The ring then consists of a band of blades with homogeneous root rings at each end, the latter having a small and slightly dovetailed mate register formed on the outer faces of the proportions shown in Fig. 514.

The very small pitch of the blades, as compared with standard practice, is noticeable, the pitch sometimes being less than $\frac{1}{8}$ in. The discs on the mandril are ready to have their edges again turned to form rings of smaller diameter.

Strengthening rings are next fixed to each end of the blade ring. These strengthening rings are made of spring steel containing 0.6 per cent. of carbon. One of them is shown in Fig. 514, to a scale of twice full size, and the drawing indicates how it is attached both to the blade ring and to the conical ring, which forms its connection with the turbine disc. The strengthening ring is first formed on the circumference of a disc, and before it is parted off, its side faces are machined, as shown by the dotted lines in Fig. 514. While the disc is still in the lathe the dovetail on the blade ring is entered into the groove on the face of the strengthening ring, and the edges of the latter squeezed in, so as to grip the dovetail tightly. To effect this, a simple device, consisting essentially of a pair of hinged arms, having rollers at their ends, is carried by the tool rest of the lathe. The arms embrace the ring, and the rollers are brought towards each other to bear against the inside and outside of the dovetail joint by means of a screw. When the lathe is started and pressure maintained on the rollers, the edges of the groove are rapidly and firmly closed upon the dovetail, and to all intents and purposes the two rings are then homogeneous.

The outboard strengthening ring is the first to be formed, and when the blade ring is secured to it the composite ring is then parted off. The other strengthening ring is then formed, and after having the composite ring inserted and fixed, this also is parted from its disc. We then have a blade ring complete except for the fine radial fins on the exterior circumference, which are provided for afterwards. The conical double-bulb-headed ring, connecting the blade ring to the turbine disc, see Fig. 509, Plate XXIV., is turned from the edge of a steel disc, but before it is parted off the blade ring is closed on to it by rollers in the manner already described, and the rectangular steel seating ring, which is caulked into a recess in a turbine disc, is also closed on to it in the same way. The conical ring is then severed from its parent disc.

The complete element, consisting of blades, strengthening ring,

conical ring, and seating ring, is then lightly chucked in the lathe, and a groove is turned round the outside of each of the strengthening rings. Into each groove a U-shaped section of extremely thin nickel is inserted, and secured by a piece of wire rolled into it (see Fig. 509). Finally, a finishing cut over the nickel fins, and on the internal points opposite the corresponding fins of the next smaller blade ring, completes the process. The seating ring is fitted into its recess in the turbine disc by means of a piece of iron wire caulked in on one side of it, as shown to the left, in Fig. 509. The ends of this wire are left slightly turned up, so that they can be gripped by a pair of pliers in case it is necessary to extract the ring.

Following on the description of the blading system, it will be convenient to deal with the dummy packings. The dummies, as seen in Fig. 516, page 354, are flat annular discs, in pairs, each annulus being attached either to the turbine disc or the fixed facing by a conical expansion ring. Between the inner pair and the next the by-pass steam for overloads is admitted to the blading. Fig. 510, Plate XXIV., which is 2.4 times full size, shows parts of the intermediate and outer pair of dummies. The fixed and running dummies are of practically identical mass and design, and so arranged that they must heat uniformly and have no relative displacement due to expansion. They are all of steel, having annular grooves machined in the lathe. The first two pair of dummies, counting from the centre, have interlocking grooves of the form illustrated in the lower half of Fig. 510. The grooves of the outermost pair do not interlock more than is necessary for the purpose of balancing end pressure. Each rib of the fixed dummies has on its outer face a small groove. Into this a folded strip of very thin nickel is lightly driven by a punch carried on a light adjustable wooden radius rod, universally jointed to a plug in the centre of the disc. The punch is then replaced by a sharp-edged roller, which divides the outstanding edges of the nickel. The thin nickel caulking wire is then tapped into position, and a flat roller, carried by the radius rod, runs over the strip, bending over the edges, and smoothing everything down nicely.

The glands adopted to secure steam tightness where the shaft passes through the turbine casing have to be packed against very

high pressures, and the steam must accordingly be wire-drawn at a very great number of constrictions if leakage losses are to be kept within reasonable limits. This has been very successfully accomplished by Mr. Ljungström. A single constituent of this gland is represented in Fig. 519, while Fig. 520 gives an excellent idea of the small dimensions of the packing.

The enlarged detail of an element of the shaft packing, consisting of one fixed ring and one running ring, is shown in Fig. 519, to a scale of twice full size. The rings are turned out of solid steel,

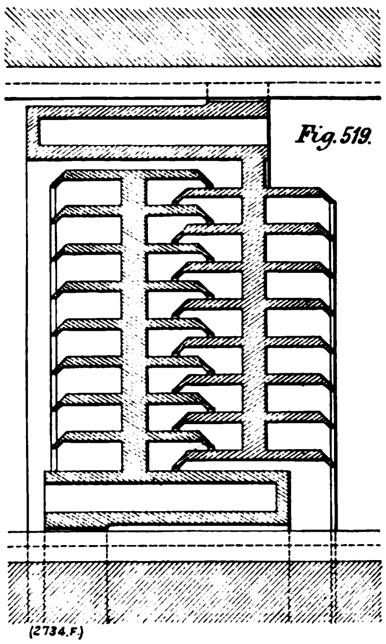


Fig. 520.

Figs. 519 and 520. Gland for Ljungström Turbine.

and are designed to expand freely in all directions without distortion. They are mounted on feathers, the fixed ones on the side disc. There are no less than 158 constrictions for the steam, although the packing occupies less than $3\frac{1}{4}$ in. of the shaft. The circumferential labyrinth walls of the rings are turned with thin edges. These edges are then rolled inwards, thus forming minute internal conical flanges, as seen in Fig. 519. When first made these flanges exactly fit the walls of the next rings, which they embrace; but a few minutes' running suffices to wear a minute working clearance, which is maintained unaltered afterwards. Actual tests showed the leakage steam from both packings of the 1000-kw. machine amounted to only about

110 lb. per hour. What steam does pass does not find its way into the generator or the engine room, but is led into a chamber, in the side discs, around the packing, whence it is taken to a feed-water heater or elsewhere by a special pipe.

Not only the shaft packing, but also all other labyrinth packings in the turbine, adjust their own clearances in the same way. Owing to the thinness of the fins, no dangerous heating occurs in case of contact, even should the running parts shift out of centre. At the same time such an occurrence as this is very carefully guarded against by the self-contained arrangement of the whole plant, its great stiffness and its symmetric form affording temperature differences the least possible chance of altering the central position to which the bearings are originally adjusted.

It is stated that the clearances of the labyrinth edges at different radii of the 1000-kw. turbine from the shaft packing outwards to the larger blade rings are easily maintained at 0.004 in. to 0.010 in., even with the high superheat used. The high pressures being confined to the inner parts, which are of small radius, the leakage as calculated comes out at a very moderate figure, even when using steam of very high pressure.

A section through the stop valve is given in Fig. 521, page 362. To start the turbine the handle at the top is screwed back a few turns, and the steam then raises the disc valve from its seat, and equalises the pressure above and below it. The double-beat piston valve above is a sufficiently easy fit in its sleeve to permit of a small amount of leakage past it, and this is quite sufficient to warm up the small mass of the turbine within a few minutes. The oil pump in the oil box at the end of the turbine is then worked by hand for a few strokes until the oil pressure is sufficient to raise the piston on the valve spindle against the downward pressure of its spring. The turbine then starts, and within about ten minutes from the commencement of warming up may be running at its full speed of 3000 revolutions per minute.

When starting there is, of course, no reason why the two sides of the turbine should drive their respective alternators at precisely the same speed, and, as a matter of fact, they do not do so. But when the speed rises to about 1400 revolutions the exciting current becomes strong enough to cause the two generators automatically

to synchronise themselves, and thenceforward to run electrically

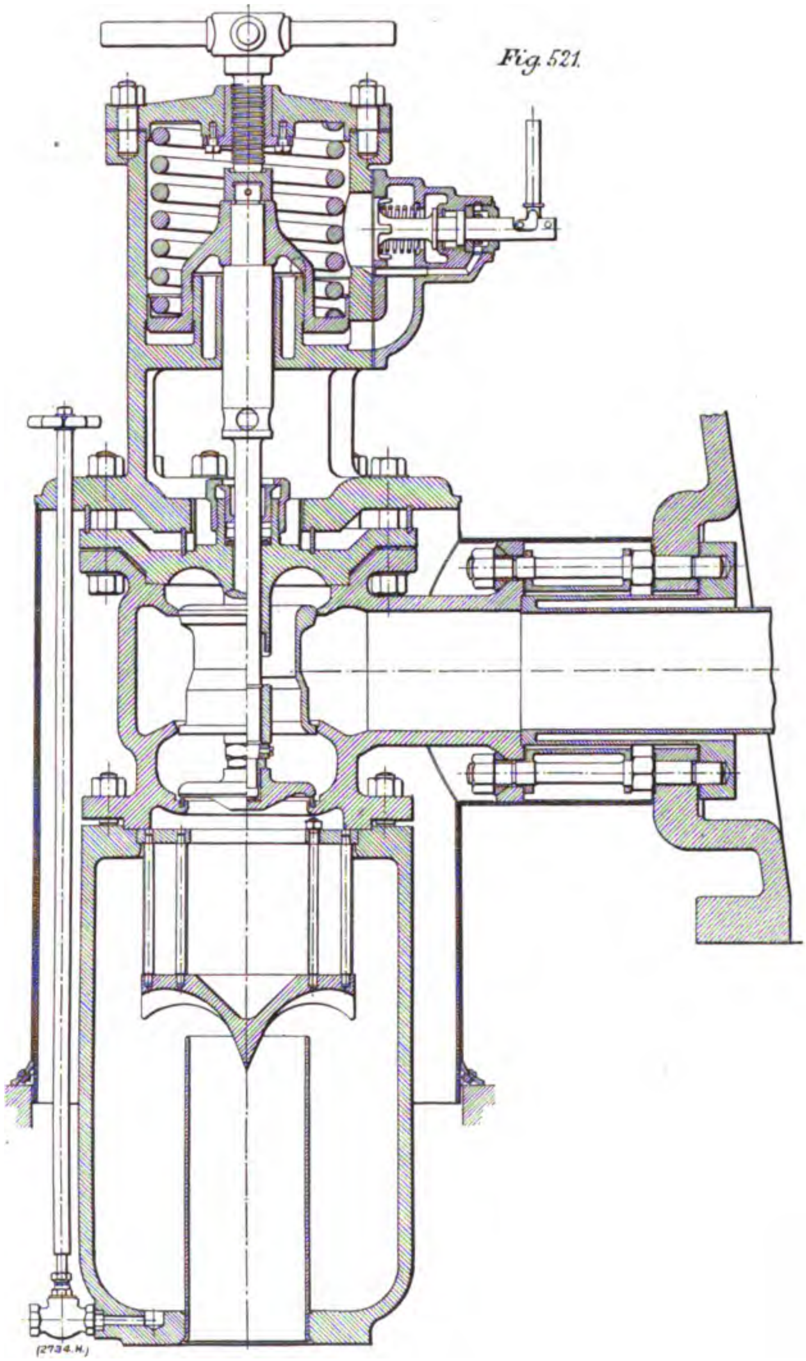


Fig. 521. Throttle Valve for Ljungström Steam Turbine.

as one machine. The fields of the two machines are connected in series, so that a single field rheostat only is needed, and in

ordinary practice no more switches or instruments would be required than are necessary for the control of an ordinary generator.

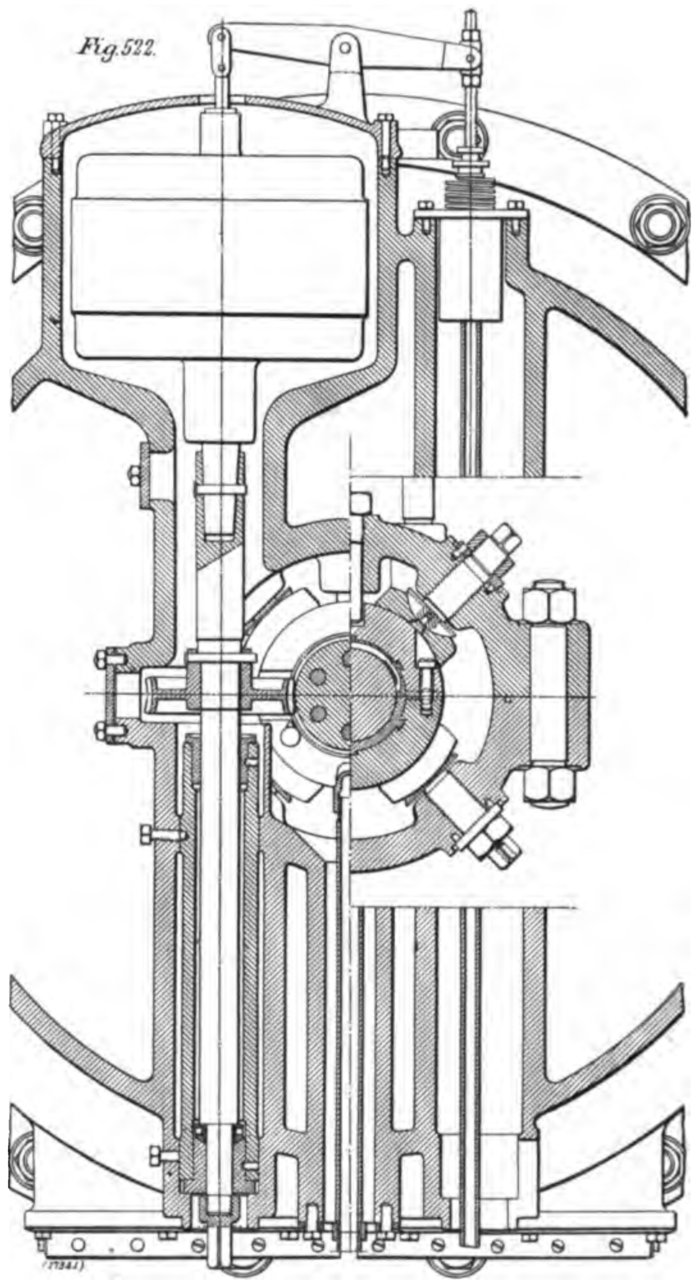


Fig. 522. Governor for Ljungström Turbine.

Should either of the generators become disabled, provision is made for bolting it rigidly to the casing, so that, if necessary, the other generator alone may be used. The result would be a reduction

of output by one half, and higher steam consumption per unit, but, the frequency and voltage being normal, an emergency supply could be maintained.

Referring again to Fig. 521, the emergency trip lever, and the valve connected to it, which releases the oil pressure and causes the main valve to be closed, are at the top to the right. The tension cable, already described as causing the stoppage of the turbine if it is slackened by any cause, such as too high speed or unduly heated bearings, passes through the small hole in the toe of the emergency lever. By turning down this lever into a horizontal position the oil pressure is released and the turbine stopped. This method of stopping can be used whenever desired, as it obviously interferes in no way with the adjustment of the cable. Another point worthy of note is the attachment of the pipe leading from the stop valve to the turbine. This is shown on the right, and it is formed by oxy-acetylene welding. Its peculiar design is intended to prevent expansion troubles from the high superheats. The simple circular form of the valves and casing is maintained with the same object.

Fig. 522 is a vertical section through the end of one of the generator casings, showing the type of bearing and governor used. The governor is driven by means of a worm on the generator shaft, and is of the Chorlton-Whitehead type described on page 262, *ante*. It controls the stop valve by means of an oil relay. The bearings, which are shown to a larger scale in Fig. 523, page 365, are short for their diameter, and consist of cast-iron steps, turned cylindrically on the outside, and lined with white metal. The two halves are registered by two long pins fitting in holes drilled along the sides of the joint when the halves are cramped together. The set screws holding the half steps together prevent the pins from working out endways. The pair of steps is located centrally by four pads fastened on to the spherical heads of screws which project through the housing, and form a means of adjustment for the position of the shaft. It will be noticed that the bearing caps are secured by taper-fitting bolts, with a nut at each end. This type of bolt is used throughout the turbine, all the casing bolts being thus made. It is claimed that they are very cheaply made and fitted, and when in position they serve to locate the

parts they hold together with absolute accuracy. They are easily withdrawn by tightening one of the nuts.

The oil supply is led to the bearing by a pipe entering the bottom step. This pipe is placed inside the discharge pipe from the bearing, as shown in Fig. 516. The incoming oil passes through holes machined in the bottom step, and emerges at the horizontal diameter at one end, whence it flows to the other end along clearances provided for the purpose in the top step. The spent oil returns to the pump from the bottom of the bearing by the outer of the two concentric pipes shown. There is no water cooling of the bearings, the oil alone being relied on to carry away the heat. The circulation of oil amounts to a pint per second for each bearing.

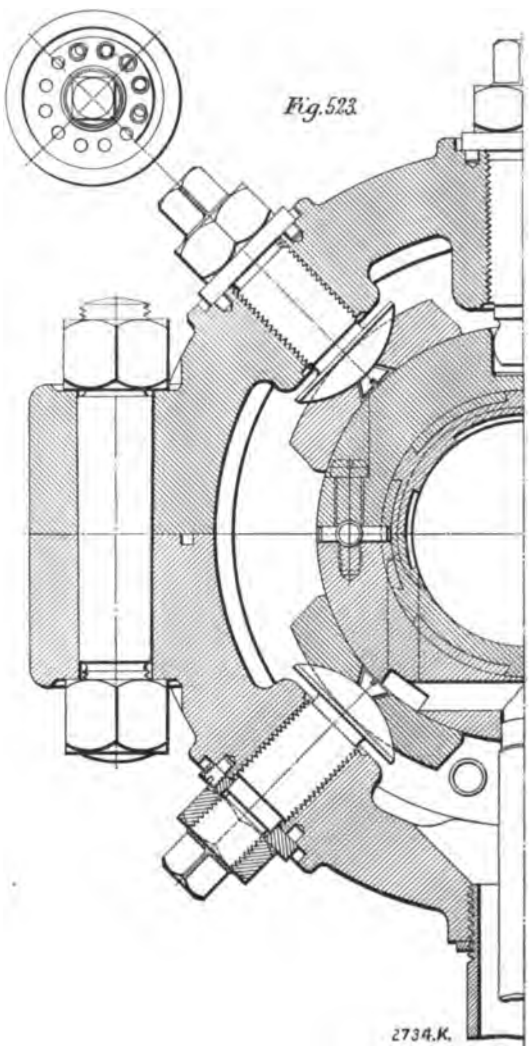


Fig. 523. Main Bearing.

The oil pump is of the gear type, and is driven directly by the governor spindle. It maintains a pressure of about 34 lb. per sq. in., but this is lowered to about 15 lb. per sq. in. for the bearings by the use of a reducing valve. The higher pressure is maintained beneath the piston controlling the throttle valve, this pressure being regulated by a slide valve operated by the governor. Variations of oil pressure thus vary the position of the throttle valve, and this, in turn, by means of a floating lever and side shaft, reacts upon the oil system, so as to maintain a constant relationship between the position of the stop valve and that of the

governor. On a full load test, with steam supplied at 130 lb. below the governor valve, and at a temperature of 664 deg. Fahr., and with an absolute pressure of 1.25 in. of mercury in the exhaust pipe, a consumption of 8 lb. per brake horse-power was realised, corresponding to a brake efficiency ratio of 76.9 per cent.

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